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OPTIMIZATION OF THE ENERGY UTILIZATION IN A LARGE SCALE COMMERCIAL OFFICE BUILDING'S HEATING, VENTILATING, AND AIR CONDITIONING SYSTEM

A Thesis

Presented for the

Master of Science

Degree

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ABSTRACT

In the Summer of 1981 the International Business Machines Corporation's Real Estate and Construction Division will complete the construction of a million square foot, 43 story office building in New York City, New York. The building's Heating, Ventilating, and Air Conditioning (HVAC) system represents the current state of the art in HVAC systems designs. This thesis presents an a priori study of the building's operation to determine its optimum operating potential.

The system has well over 25 valid configurations. This work addresses the optimum operation of each configuration, and other factors which must be considered in selecting the optimal configuration. All the results presented in this thesis are based upon a detailed steady state simulation of the building's HVAC system. A previous thesis by Marcev (2) details the development of the system simulation.

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INTRODUCTION

The use of digital simulations to study the design, operation, and economics of large scale systems has become widespread in the chemical process industries (1). This acceptance of mathematical modeling as an effective optimization tool has not been as general in the Heating, Ventilating, and Air Conditioning (HVAC) industry. Most of the simulation packages that have been developed focus primarily upon the estimation of system heating and cooling loads for the purpose of sizing a building's mechanical equipment (chillers, supply fans, ducts, etc.). System designs in the past have been focused primarily on capital cost and for the resultant simple systems there has been very little need for developing steady-state simulations that permit the operating behavior and economics to be studied. With the increasing cost of energy these inefficient designs have become economically unattractive and the trend has shifted toward systems that have multiple sources of energy available to meet the heating, cooling, and dehumidification demands of a building's occupants. This redundancy in sources has dramatically increased the operational complexity of HVAC systems, and has created a strong need for comprehensive simulations which allow the operating trends and economics to be studied.

The University of Tennessee's Applied Control Laboratories (ACL) has started a program for developing component mathematical models of the equipment typically found in commercial HVAC applications (2). Manufacturer's design data or the experimentally observed behavior of a piece of equipment is used to fit the corresponding model's

characteristics to the observed ones. These parametric models can then be linked together to create simulations of a system of interest. This work reports on the use of the simulation package in the a priori study of a commercial office building currently being completed in New York City (2). The building is being constructed by IBM Corporation's Real Estate and Construction Division (RECD) and the design philosophy reflects the current trend towards highly integrated HVAC system designs.

This thesis addresses the operational optimization of the building's central heating and cooling facility. In the next chapter, the pertinent system simulation literature will be discussed. The chapters to follow will:

1. Describe the building and its HVAC system.
2. Describe the organization of the central plant's operating potential and the steady-state simulations developed to describe the plant.
3. Detail the solution to the local equipment optimization problem.
4. Describe the development of the global optimization algorithms.
5. Present some case studies which illustrate the energy conservation potential of the building.

Even though the algorithms and procedures described in this work are specifically for IBM's 590 Madison Avenue office building, it is expected that the results are general enough to be applicable to other facilities.

CHAPTER I

HVAC SYSTEM SIMULATIONS

The use of the digital computer for system simulations has generally been limited to the estimation of building heating and cooling loads (3,4). These simulations (5,6,7) are used as design aids in equipment selection and are of little value in the sensitivity studies required to develop an operational optimization of any HVAC system.

Since 1971, Stoecker (8,9,10,11) in conjunction with the American Society of Heating, Ventilating, and Refrigeration Engineers (ASHRAE) has been concerned with the development of procedures for performing steady state simulations of the components and subsystems found in HVAC applications. Stoecker's work addresses the problem of model selection and simulation design. No results concerning systematic studies of these simulations have been published. Marcev (2), at The University of Tennessee, also went into great detail concerning the development of simulations for optimization studies, but presented no results from such studies.

Most of the system studies presented in the HVAC literature have been based upon simplistic models. For example, one study which compared the economizer (enthalpy) cycle to conventional heating and cooling used a chilled water power consumption model of 0.75 killowatt hours per ton of chilled water load (12). This kind of model, though useful for rough estimation of the power consumption of various standard strategies, is inadequate for use in development of local optimization strategies or in studies of the subtle interactions between building subsystems. It is understandable that little has been published about simulation results

because the configuration for each HVAC application is unique. There is also a formidable amount of heuristic knowledge (13,14,15,16) about the performance of HVAC systems and subsystems and these simulation studies would probably only provide confirmation of most of this knowledge. But the digital system simulation provides the operator of an HVAC system the tools needed to evaluate heuristic procedures in the light of the current and future (predicted) cost of energy. New rules of thumb may emerge from such studies or analytical expressions may be arrived at which accurately describe the interactions between subsystems, local constraints, and utility consumption.

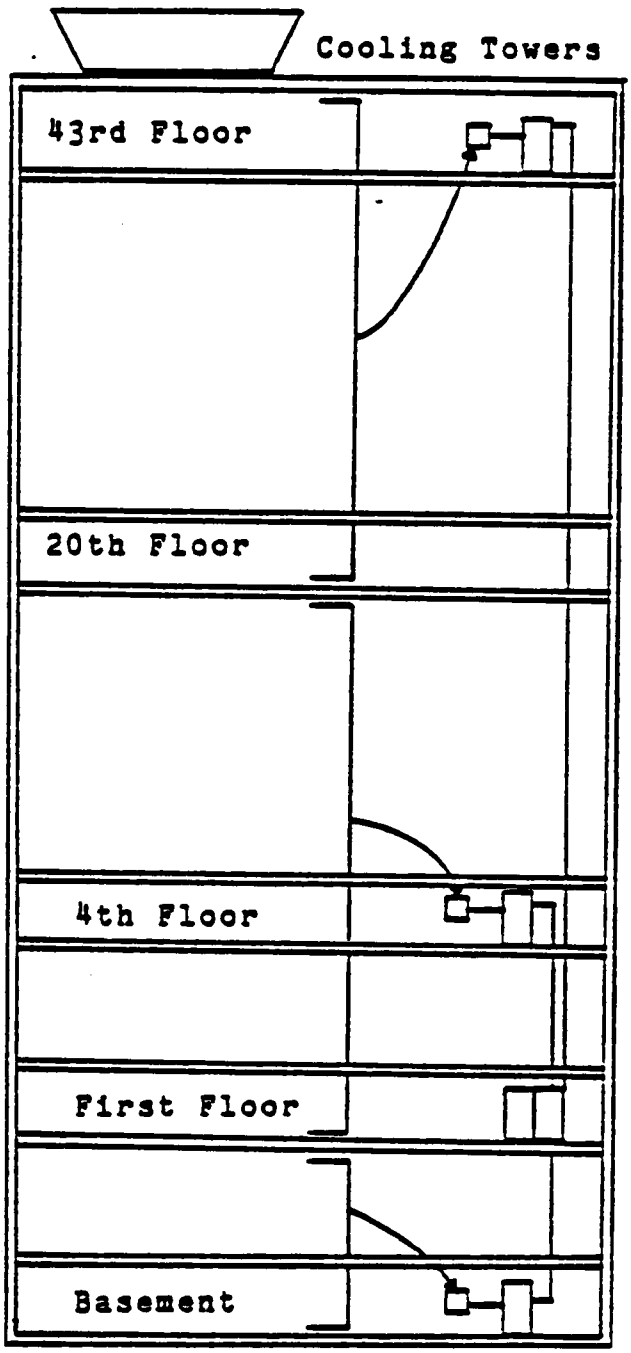
Since it has been suggested that as much as three percent of the total energy consumption of the U.S. could be saved by the proper maintenance, operation, and repair of existing HVAC systems (17), it is essential for HVAC personnel to understand system operation. The system or subsystem simulation can help provide this understanding. Through the use of a digital simulation, this work will study a building subsystem and compare optimization and operation results with some of those results which are available in the HVAC literature (18,19,20).

CHAPTER II

BUILDING DESCRIPTION

The HVAC system of this building is unique in many ways and represents a departure from traditional design concepts. One of the most significant and novel features is the emphasis that has been placed upon instrumentation and computerization throughout the building. This high degree of automation provides the capability for monitoring and controlling the building's energy usage from a central location. The computers used range from an IBM 370 that is responsible for coordination of the total building operation to small eight bit microprocessors that are controlling office temperatures. All of the building's 2000 plus processors are linked together by a bidirectional communications network. The data acquisition and reduction required by the global optimization supervisor is provided by three IBM Series/1 (S/1) minicomputers. One of these S/1's is dedicated to coordinating and optimizing the performance of the central chilled and hot water facility. The other two are responsible for the control of the operation of the air handlers and the associated office space conditions in the upper and lower halves of the building. A sketch of the building with the locations of the computers is shown in Figure 1.1.

The building is essentially an isoceles triangle (Figure 1.2) with faces directed toward the east, northwest, and southwest. The building space is divided into the interior and perimeter zones. There are three perimeter zones made up of offices adjoining each of the three



Air handler and secondary water systems. Remote series/1 with CCM interface for control of air handlers, IZM's and PZM's in upper half of building.

Air handler and secondary water system. Remote series/1 with CCM interface for air handlers, IZM's and PZM's in lower half of building.

Central series/1 mini-computer room. Provides operator interface, selects optimum operational strategies and maintains data base.

Refrigeration machine room. Remote series/1 with CCM interface for water systems and minor air handlers.

Figure 1.1. Location of major HVAC equipment and process control computers in the 590 Madison Avenue building.

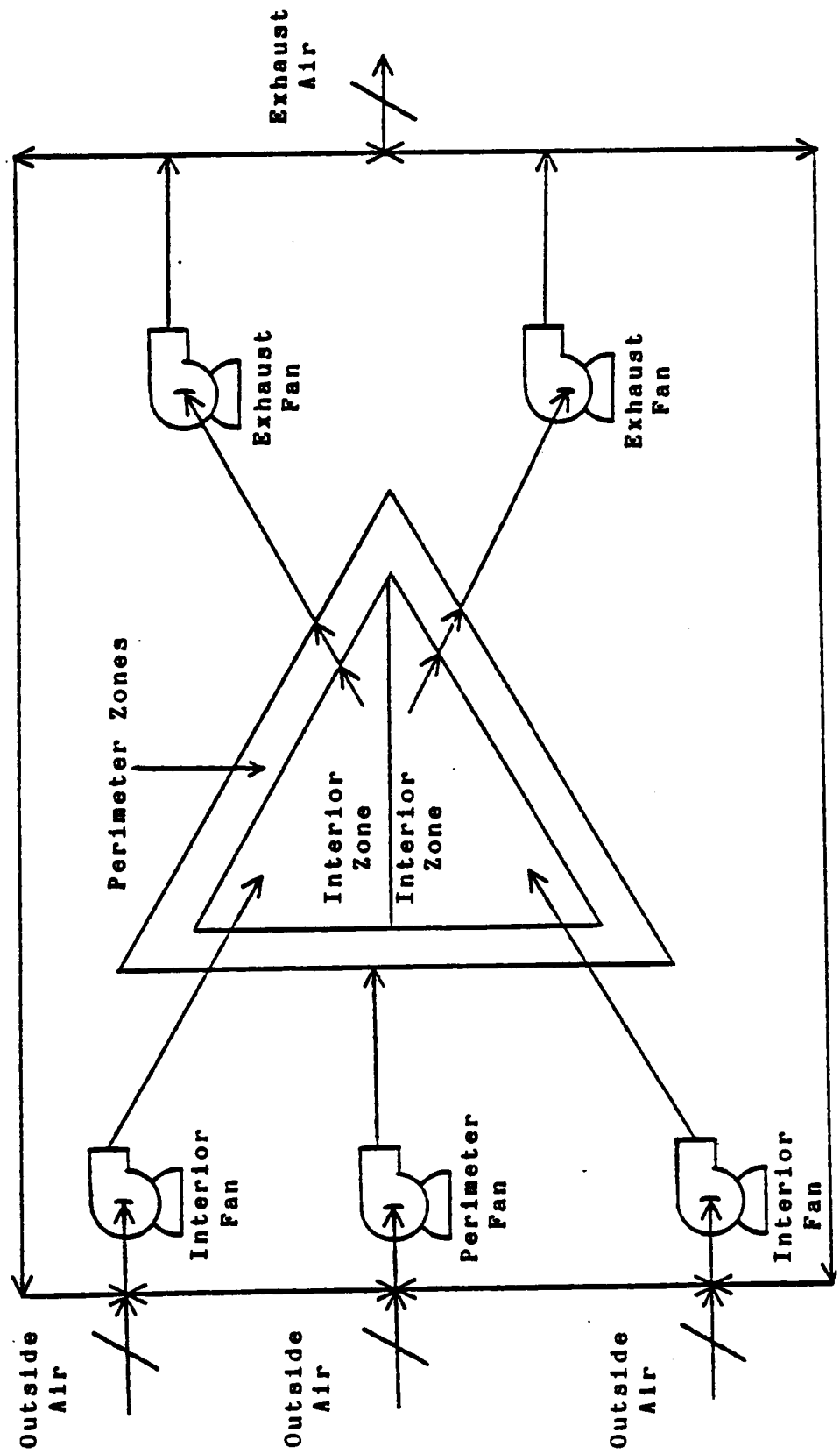


Figure 1.2 IBM's 590 building ventilation system description.

faces respectively and two interior zones, each containing half of the building space for both the top and bottom sections of the building. Offices in the perimeter zones have a wall that is in thermal contact with the outside and effectively isolate the interior zones from the ambient environment. Consequently, the interior offices have a net heat gain throughout the year due to occupancy (lighting loads, equipment loads, etc.) except under the most extreme of outside temperatures. The cooling, ventilation, and dehumidification requirements of the interior are met by a variable air volume (VAV) system. Each floor has two interior zones. The temperature of each of these zones is controlled by regulating the air flow into the zone by manipulating the variable air damper associated with the zone. The microprocessor room thermostat maintains a filtered valve of the zone's damper position and returns it to the local S/1 upon request along the data highway. The loading information, gathered from the building's 400 interior thermostats, is used by the S/1 in optimizing the operation of the interior air handlers.

The perimeter office conditions are individually maintained by a two pipe induction unit. Each unit consists of a small air-water heat exchanger through which room air is circulated by a venturi type induction unit using high pressure supply air. The unit is controlled by a small microprocessor based thermostat which regulates the water flow (note that the water may be either hot or cool) to the heat exchanger to control the local office temperature. Each processor maintains a record of the fraction of the time its control valve is in the open position. This information is available to the local S/1 for determining the

relative load on each of over 2000 perimeter offices. This relative loading information is used in optimizing the performance of the perimeter air handler.

The utility of using the relative load on each office as an energy management tool is enhanced by the fact that the three perimeter faces are almost completely independent of one another. This permits each face to be configured to meet its needs using the most cost effective combination of resources. The faces are not totally independent because the cooling and dehumidification of the primary air is provided centrally. The required temperature of the primary air is established by the perimeter face needing the coolest air or by dehumidification constraints. Beyond the supply fan the ductwork branches to the three faces, and each face has its individual reheat coil to elevate the temperature of the air to meet the requirements of that particular face.

Each face has its own secondary water system. The temperature of the water is maintained through heat exchangers using chilled or hot water supplied by the primary water subsystem. The set point temperature for the secondary water systems and the temperatures of the air from the reheat coils are established by using the relative loading information from the individual room thermostats. Feedback strategies attempt to coordinate the operation to minimize the simultaneous heating and cooling of the zones by the two subsystems.

Though the optimization of the air handlers and the secondary water subsystem is important, the optimization of the central refrigeration and heating facility has been determined to be much more critical

for the economic operation of the building. For this reason this work presents the optimization studies that have been performed on the primary water subsystem.

Primary Water Subsystem

The Primary Water Subsystem (PWSS) consists of all the equipment, pumps, piping, and valving required to supply the chilled and hot water needs of the air handlers, the secondary water subsystems, and the computer rooms of the building. The mechanical equipment includes three centrifugal chillers, a four cell forced draft cooling tower, and the variable speed pumps of the three pumping networks (cooling tower, chilled and hot water). Two of the centrifugal chillers may be operated as heat pumps with their condensors connected directly to the hot water network. These two electric motor driven chillers have cooling capacities of 1250 and 250 tons. The third chiller has a steam turbine drive with a capacity of 1250 tons.

When economic or loading conditions make the operation of the heat pumps undesirable or unfeasible, a steam to water heat exchange is available for the production of hot water. The steam required for the production of hot water is supplied by the local utility company. Chilled water can be provided either by the chillers or by the cooling tower, after first passing the tower water through a strainer, when outside conditions are appropriate. Thus, there are two primary heat sources available for hot water production and two mechanical sources available for chiller water production.

CHAPTER III

ORGANIZATION OF THE PWSS OPERATING POTENTIAL

The first step in developing an optimization program for the PWSS is to organize the subsystem's operating configurations into a logical framework. This is accomplished by grouping configurations together that have common features. The most natural classification is to define major operating modes that group configurations according to their primary energy sources; that is, which major pieces of equipment are running and how are they connected. Submodes are then used to define which specific refrigeration machine or machines are operational. Therefore by giving a mode and submode designation, one indicates which equipment is running and the valving required in the pumping network.

The PWSS's forty-one operating configurations can be readily organized into the framework of eight major modes. The mode definitions are provided in Table 3.1 and the submode definitions in Table 3.2. In six modes [0,1,4,5,6, and 7] the subsystem is simultaneously supplying heating and cooling. Two of these six modes [0 and 4] use steam to produce hot water and when no heating is needed these two modes are renamed to create the final two operating modes [2 and 3]. In the other four of these modes [1,5,6, and 7], a heat pump is the primary heating source. The mode labeled Heat Pump [7] requires the special conditions of having the heating and cooling loads on the PWSS matched.

Chilled water is always required by the building because of the cooling required in the computer rooms. In two modes [0 and 2], the cooling tower is configured to run in parallel with the evaporators of

TABLE 3.1

DESCRIPTION OF THE MODES OF OPERATION OF THE PWSS

Mode	Description
0	Free cooling with the cooling tower, with no pay heating required.
1	Free cooling with recovered heat provided by a heat pump.
2	Free cooling with the cooling tower, with no pay heating required.
3	Mechanical Refrigeration.
4	Mechanical Refrigeration with steam heat.
5	Mechanical Refrigeration with recovered heat.
6	Partial heat recovery (with some heat rejected through cooling tower).
7	Heat pump (with cooling tower not operating).

TABLE 3.2

DESCRIPTION OF THE SUBMODES OF THE PWSS

Submode	RM-3 Small Electric	RM-3 Large Electric	RM-1 Steam
0	--	--	--
1	C		
2		C	
3	C	C	
4			C
5	C		C
6		C	C
7	C	C	C
8	H		
9		H	
10	H	H	
11	H	C	
12	H		C
13	C	H	
14		H	C
15	H	H	C
16	H	C	C
17	C	H	C

C = Chiller Operation

H = Heat Pump Operation

the heat pump. In two modes [6 and 7] a heat pump supplies all of the building's chilled and hot water needs. In the Partial Heat Recovery Mode [6] the cooling tower rejects any of the excess heat not required by the hot water users while in the Heat Pump Mode [7] all of the hot water produced by the heat pumps must be used by the hot water users. Finally, in three of the eight operating modes, [3, 4, and 5], the cooling tower is connected directly to the condensors of the chillers, which in the past was the typical mode of operation.

The mode-submode scheme permits the configuration of the tower, the refrigeration machines, and other equipment to be readily visualized. This is important not only to the building operator but also in the development of optimization strategies. It aids in discerning similarities between modes so that generalizations can be made in solving local optimization problems. One such generalization to be shown is that of optimizing the chiller/cooling tower combination which appears in 70% of the configurations chosen for study. Another is that the chilled and hot water production facilities can be optimized independently when steam is the primary heating source.

Thus it can be said that the mode-submode organizing framework clarifies the operating potential of the PWSS and brings the maze of configurations into clear focus.

Simulation Development

With the subsystem clearly organized, the next step in solving the optimization problem is the development of steady-state simulations of key configurations. Twenty of the most representative configurations,

which cover the full spectrum of operating modes, were selected for study.

The simulations were created by linking together detailed component models of the mechanical equipment, pumps and piping networks. An input data set is provided to the simulations which specified the values of a set of system variables. The simulations then solve the material and energy balances and return in a compact report the remainder of the system variables, utility requirements of all equipment, stream flows, stream temperatures, equipment loads, and error codes. The error codes indicate whether any equipment constraints would be violated if a particular configuration were implemented with the input data set.

The simulations permit the operation of the PWSS to be studied over the full range of system loads that are expected. Extensive execution of the simulations provided insight into the sensitivity of the modes of the PWSS to the various system variables, whether these variables are independent or manipulated. This aided in the development of the local optimization strategies that are to follow.

CHAPTER IV

PRIMARY WATER SUBSYSTEM OPTIMIZATION

Justification for Independent Optimization of the Subsystem

The object of optimization of a HVAC system is to minimize the purchased cost of utilities. To perform this minimization it is desirable to have a simple functional relationship, the objective function, which describes utility consumption as a function of system variables. In a system such as this, with many variables and complex relationships between the variables, the formulation of a simple objective function is difficult. If the system can be decomposed into a set of smaller subsystems which may be optimized independently, then the problem is greatly simplified.

System decomposition. The HVAC system of 590 is comprised of three physically distinct subsystems. These are the primary air, the primary water, and the secondary water subsystems. By use of an optimization technique called splitting (21), it is proposed that the primary water system may be optimized independently of the other subsystems.

"Splitting" describes a technique which is used to decompose complex optimization problems into sets of smaller problems. The separation is valid "as long as the split is made at a point where the interconnecting streams are fixed in value" (21). Systems may be decomposed even if no fixed links are present if "certain connections between subsystems are temporarily regarded as being specified by constants" (21). To use splitting one must be able to identify a stream which interconnects subsystems and justify treating the conditions of that stream as constant. It is also presumed that the operation of the connecting link is optimal.

Figure 4.1 shows a block diagram of the overall HVAC system of 590. The primary water subsystem includes the chiller/cooling tower combination. This subsystem is interconnected to the rest of the system via the two pumping networks. If the streams of the pumping networks are operated optimally and can be regarded as temporarily fixed in value then the primary water system may be separated and optimized independently of the rest of the system over the interval in which the streams are constant. Since the dynamics of the connecting streams are ultimately tied to the dynamics of the building space through the primary air and secondary water subsystems, one must ask if it is adequate to assume the connecting streams are constant. To answer this question, the optimal operation of the pumping networks must first be considered.

Pumping network optimization. For splitting to be valid pumping networks must be operated optimally. In Figure 4.2 is a control system which will optimize the operation of the pumping networks. The optimization goal is to maximize (minimize) the chilled (hot) water supply temperature provided by PWSS. The Valve Position Controller's (VPC) task is to find this temperature. The optimum temperature will supply just enough cooling (or heating) to meet all user's loads while minimizing the pressure drop across the user's control valves. In Figure 4.2 the controller outputs from all users are fed to a device which selects the highest input signal (HS) and feeds the signal to the VPC. The VPC will adjust the supply temperature set point to PWSS to maintain one input signal about 90 percent of its maximum value. All users are satisfied and pressure losses due to throttling at the control are minimized. The VPC attempts to find the optimum steady-state supply temperature and should be a slow integral controller. The controller is

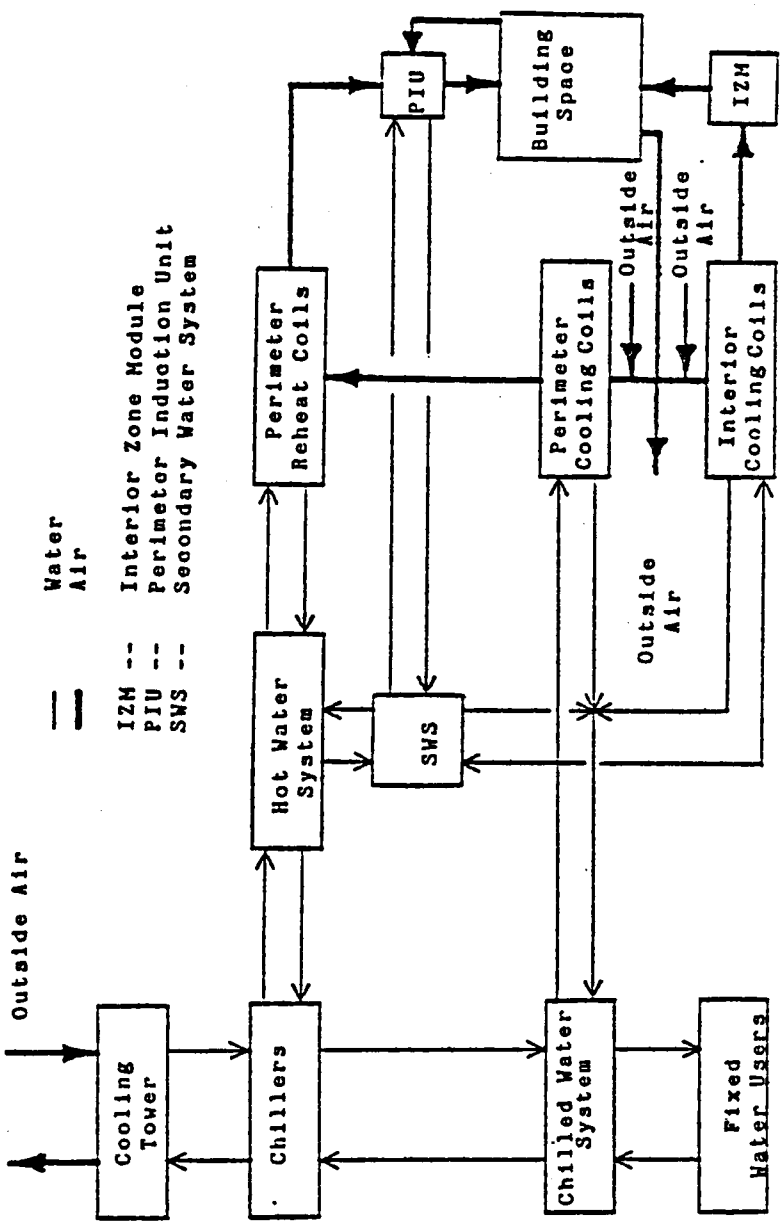


Figure 4.1. Interactions between air and water systems in 590 Madison Avenue's building HVAC system.

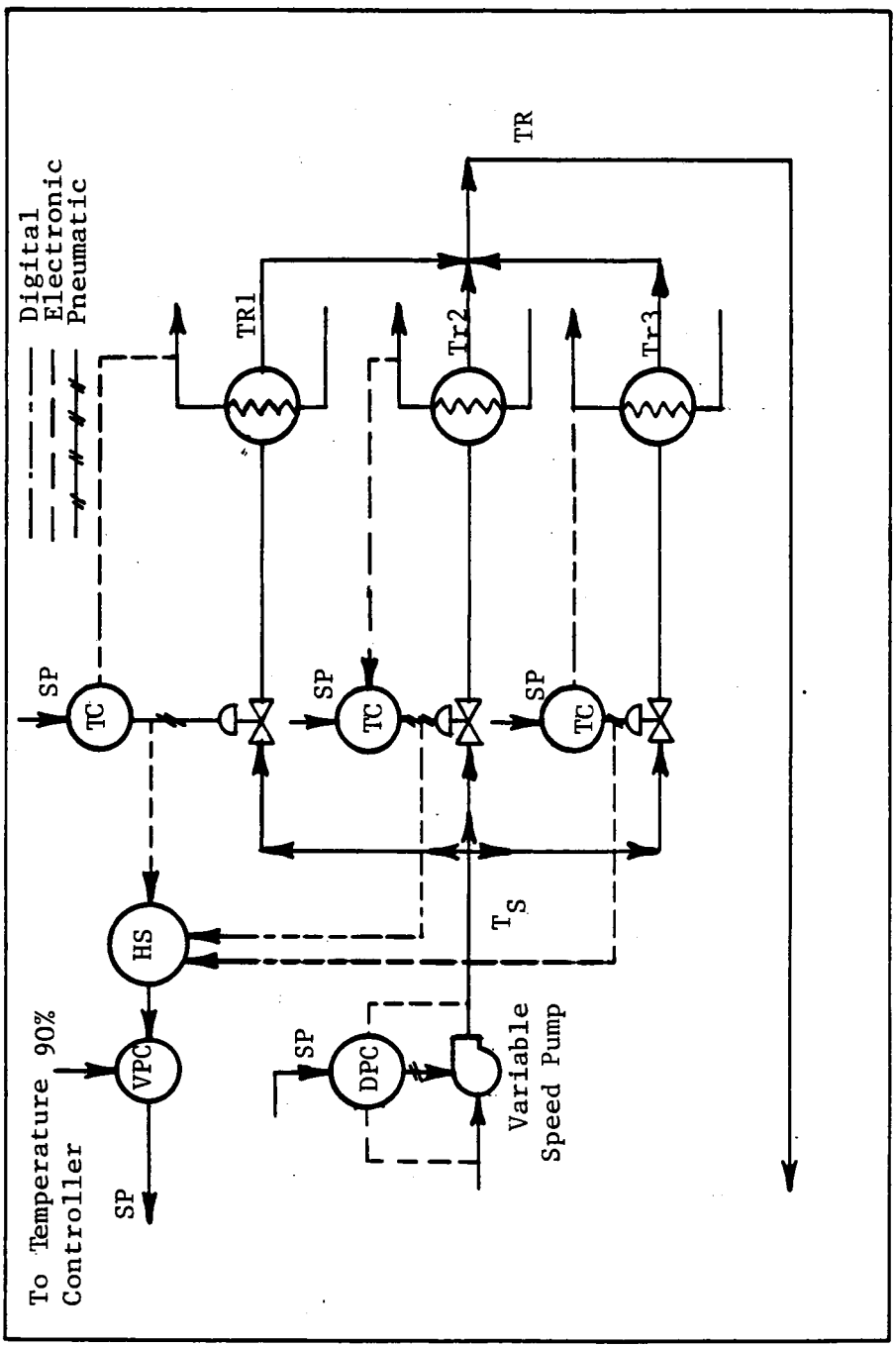


Figure 4.2. Primary hot or chilled water system feedback control scheme to continuously optimize the supply temperature and the variable speed pump.

reverse acting for chilled water and is direct acting for hot water. (A direct acting controller's output increases as the control variable increases.)

The total flow through the networks is controlled by a Pressure Drop Controller (DPC). This is an analog controller. The set point to this controller should maintain the total flow high enough for good heat transfer at all of the user's heat exchangers.

In summary, the supply temperatures, flowrates, and loads may be regarded as essentially constant because of the building dynamics and the slow integral times of the VPC and DPC controllers. The pumping networks are also operated optimally. The PWSS is therefore separable from the other subsystems using the splitting technique.

Primary Focus of the Local Equipment Optimization

Simulations of twenty configurations were developed. Of these twenty, fourteen include the chiller-cooling tower combination. Of the other six, all but two (free cooling via the cooling tower) use the chillers configured as heat pumps. Two modes, 3 and 4, differ only in that in Mode 4 the building requires heating, and in Mode 3 it does not. Each of these modes have six submodes. Since the heating in Mode 4 is provided by steam and the hot water network is optimized via the control scheme in Figure 4.2, the optimization of Mode 3 and Mode 4 is the same. This reduces the number of independent configurations which have to be studied to thirteen. It will be shown later that in the other configurations (Modes 1.8, 1.9, 2.0, 5.11, 5.12, 6.1, 6.2) there are few significant optimization handles. Therefore, the local equipment optimization study focused on the optimization of the chiller-cooling tower equipment

combination, which is primarily Mode 3 (Mechanical Refrigeration). The optimization of the other modes is similar and in most cases simpler than the optimization of Mode 3. The aim of this chapter is to show that the selection of the appropriate set points to optimize the PWSS is relatively straightforward.

Description of Mechanical Refrigeration Modes

590 has three centrifugal chillers. Since it is assumed that all three chillers will never be running simultaneously, Mechanical Refrigeration Mode 3 has six valid configurations. ACL developed and studied all six of these submodes and all of the optimization algorithms are based upon these studies. In the study of the submodes it was assumed that the BTU load on the chillers is set by the various users and the flow is established by the optimization of Figure 4.2, and that ambient conditions established the minimum cooling water temperature from the cooling tower. The load on the chiller is composed of the supply temperature, flow, and cooling range (delta T) of the chilled water. The studies also assumed that first level control strategies will allow the optimization algorithms to set the cooling tower supply temperature and the cooling water flowrate through the cooling tower regardless of the equipment configuration. Furthermore, if two chillers are running it was assumed that the supply temperature of the electric chiller and the flow distribution between chillers can be specified. The simulations reported any equipment constraints which have to be violated to meet these specifications. ACL then developed the optimization algorithms. The results were a set of simple, easily implemented procedures which were relatively insensitive to system loads, supply temperatures, flowrates, chilled water range, and ambient

conditions. The results were also in agreement with work available in HVAC literature (13,14,15). The procedural algorithms are summarized at the end of this chapter, and the following is a detailed discussion of the studies which led to the procedural algorithms.

Discussion of the Optimization of Mode 3.2

Mode 3.2 is a submode of mechanical refrigeration. In this submode only the 1250 ton electric chiller and the cooling tower are operating to provide chilled water. Figure 4.3 shows a schematic of the submode. The figure is valid for either Mode 3 or Mode 4 because heating loads are met by steam. The schematic is typical of the three submodes where one chiller is operating. The other two submodes are:

1. Mode 3.1: only the 250 ton electric chiller is on.
2. Mode 3.4: only the 1250 ton electric chiller is on.

As has been discussed earlier, the chiller cooling tower combination occurs in fourteen of the twenty configurations studied. It is essential that the cooling tower's effect upon the subsystems energy cost be understood. In most older HVAC systems, chiller designs have led operations personnel to establish an arbitrary high minimum cooling water temperature of 75°F (13). With improved designs this limit can be relaxed and considerable improvements in energy cost realized.

The chiller models used in the simulations have no restrictions on the cooling water supply temperature. But the manufacturers recommended limit of twenty degrees temperature difference between condenser exit and chilled water temperature is followed. The simulations report violations of this constraint. 55°F was chosen as the minimum condenser inlet temperature and followed the manufacturers recommended temperature differences.

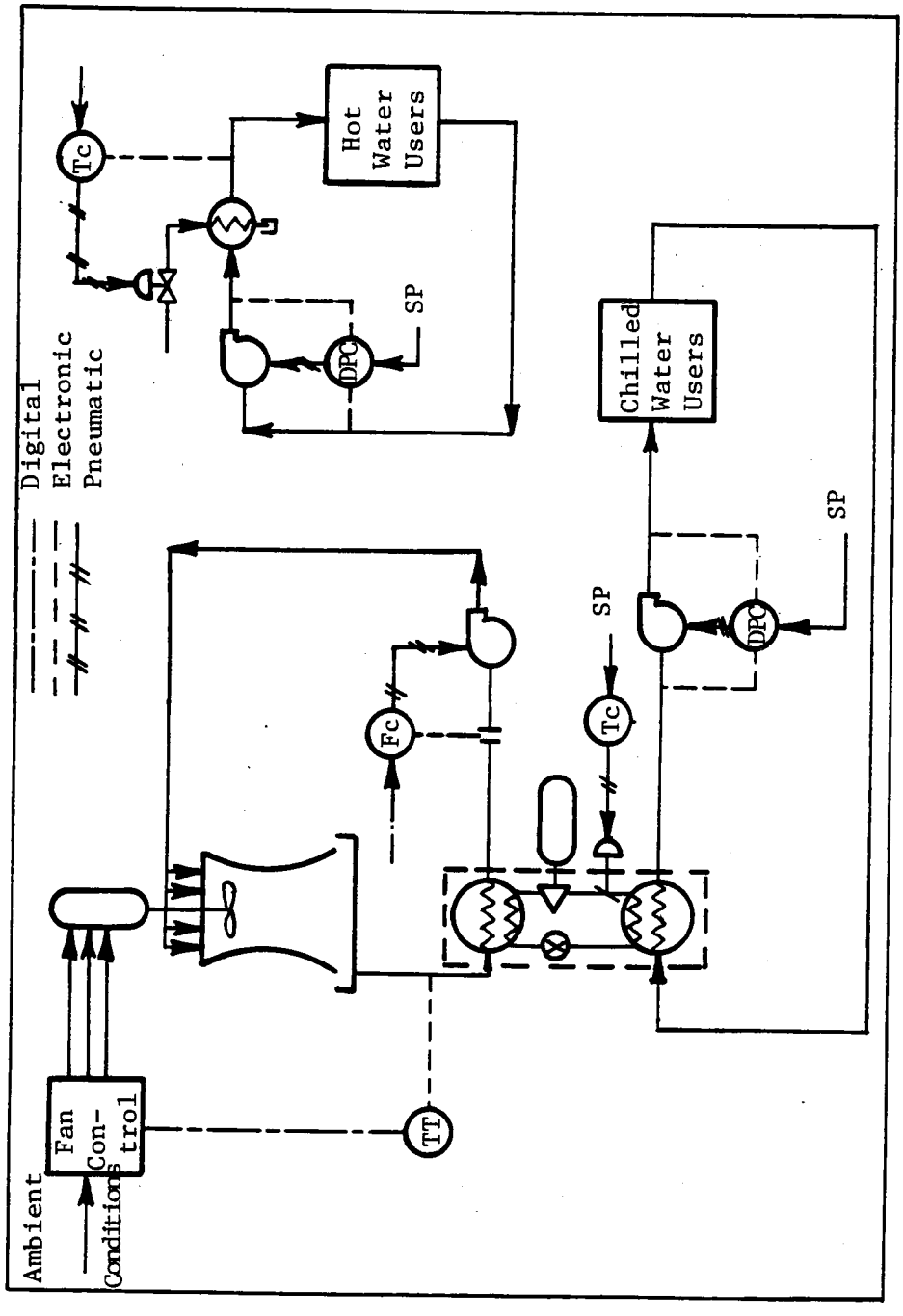


Figure 4.3. Mechanical refrigeration Mode 3.1, 3.2, or 3.4.

ACL also limited the range of chilled water supply temperatures studied to ten degrees Fahrenheit (10°F). Since it is generally agreed that 40°F is the coolest temperature required in most HVAC systems (13), this was the base temperature of the study. After considerable work it was concluded that the cooling water optimization is insensitive to the chilled water supply temperature and load for normal operations. Most of the optimization studies are presented for 44°F chilled water with a ten degree temperature drop across the chillers. The justification for this will be presented in detail by a study of Mode 3.2.

The chilled water supply temperature and flowrate are set by the feedback control scheme discussed earlier and presented in Figure 4.2. The hot water network optimization is similar. The chilled water conditions are completely specified at any instant and are independent variables to the optimization. The ambient air conditions (wet and dry bulb temperatures and relative humidity) are the other independent variables. The variables which the optimization algorithm can manipulate are:

1. Cooling water set point to the cooling tower fan control.
2. The cooling water flowrate set point to the pump controller.

Constant cooling tower approach temperature. The first set of sensitivity studies shows the effect of chilled water loads, chilled water supply temperatures, and ambient conditions on the optimum cooling tower approach temperature at a constant cooling water flowrate. (The cooling tower approach is defined as the difference between the cooling water supply and the ambient wet bulb temperature.)

Figures 4.4 to 4.11 summarize the results of these studies. Figures 4.4 to 4.6 are for a 930 ton load with a 68°F wet bulb. The

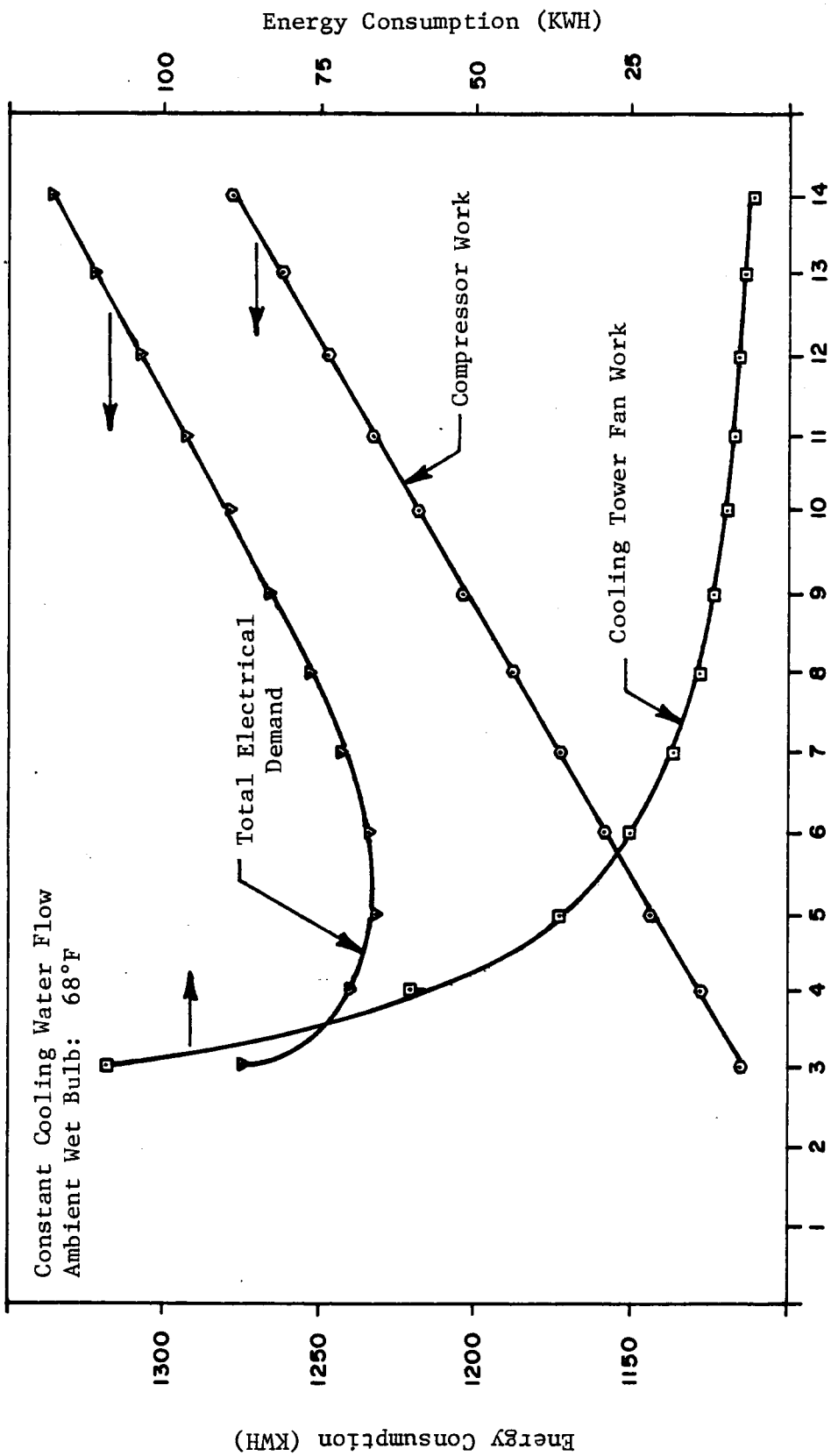


Figure 4.4. Mode 3.2 electrical demand as cooling tower approach temperature varies for 40°F chilled water and a 933 ton load.

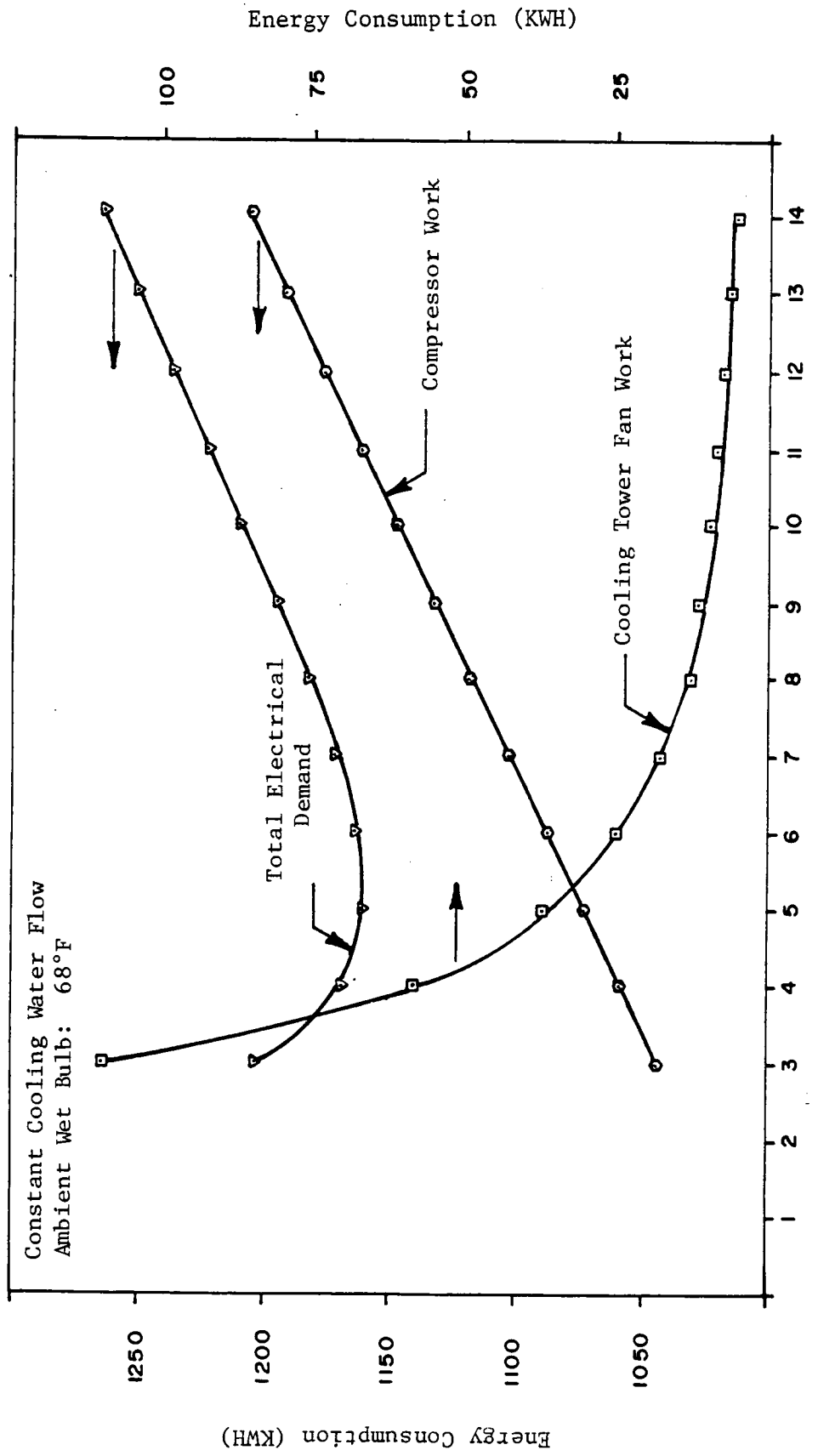


Figure 4.5. Mode 3.2 electrical demand as cooling tower approach temperature varies for 44°F chilled water and a 933 ton load.

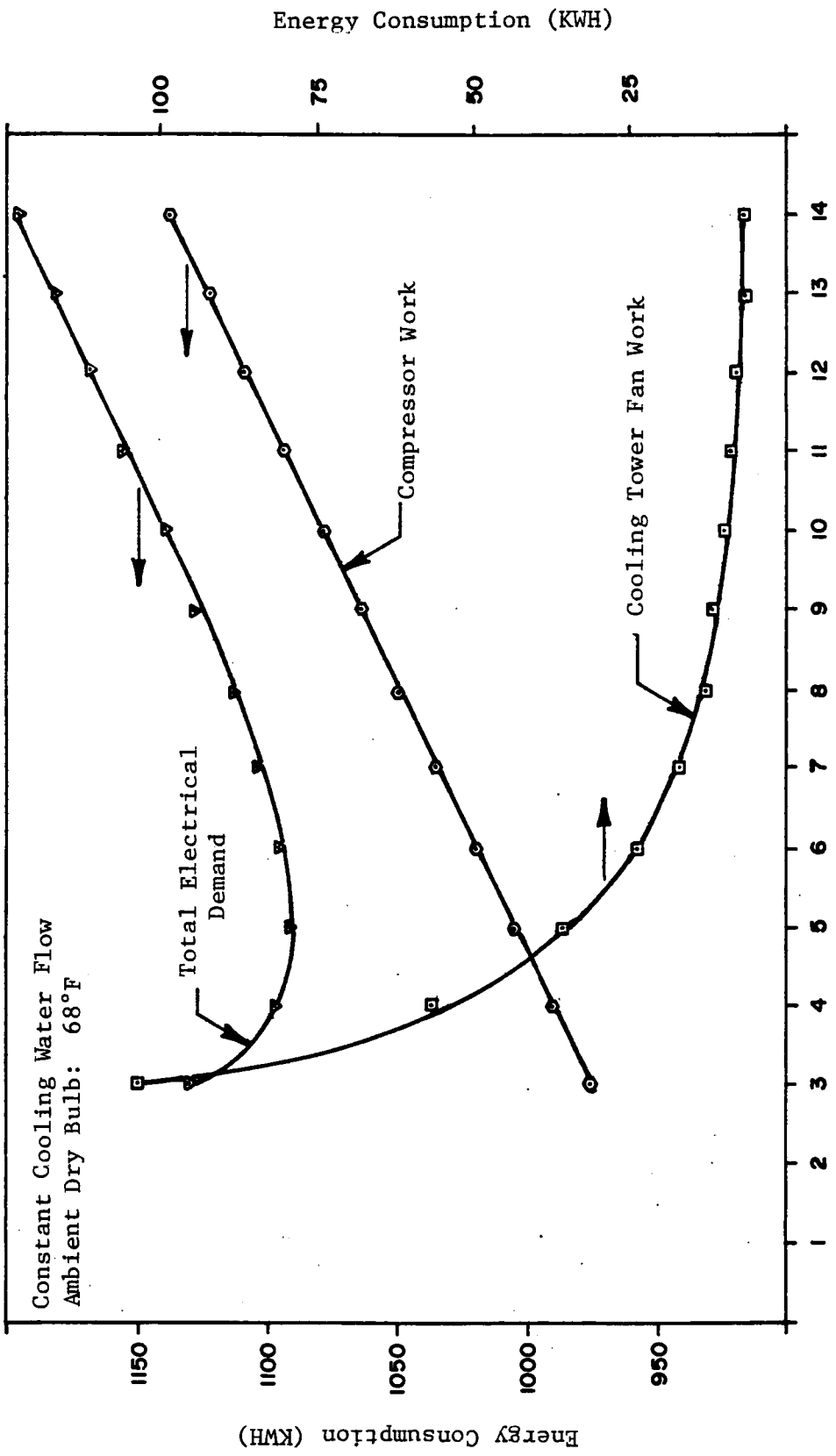


Figure 4.6. Mode 3.2 electrical demand as cooling tower temperature varies for 48°F chilled water and a 933 ton load.

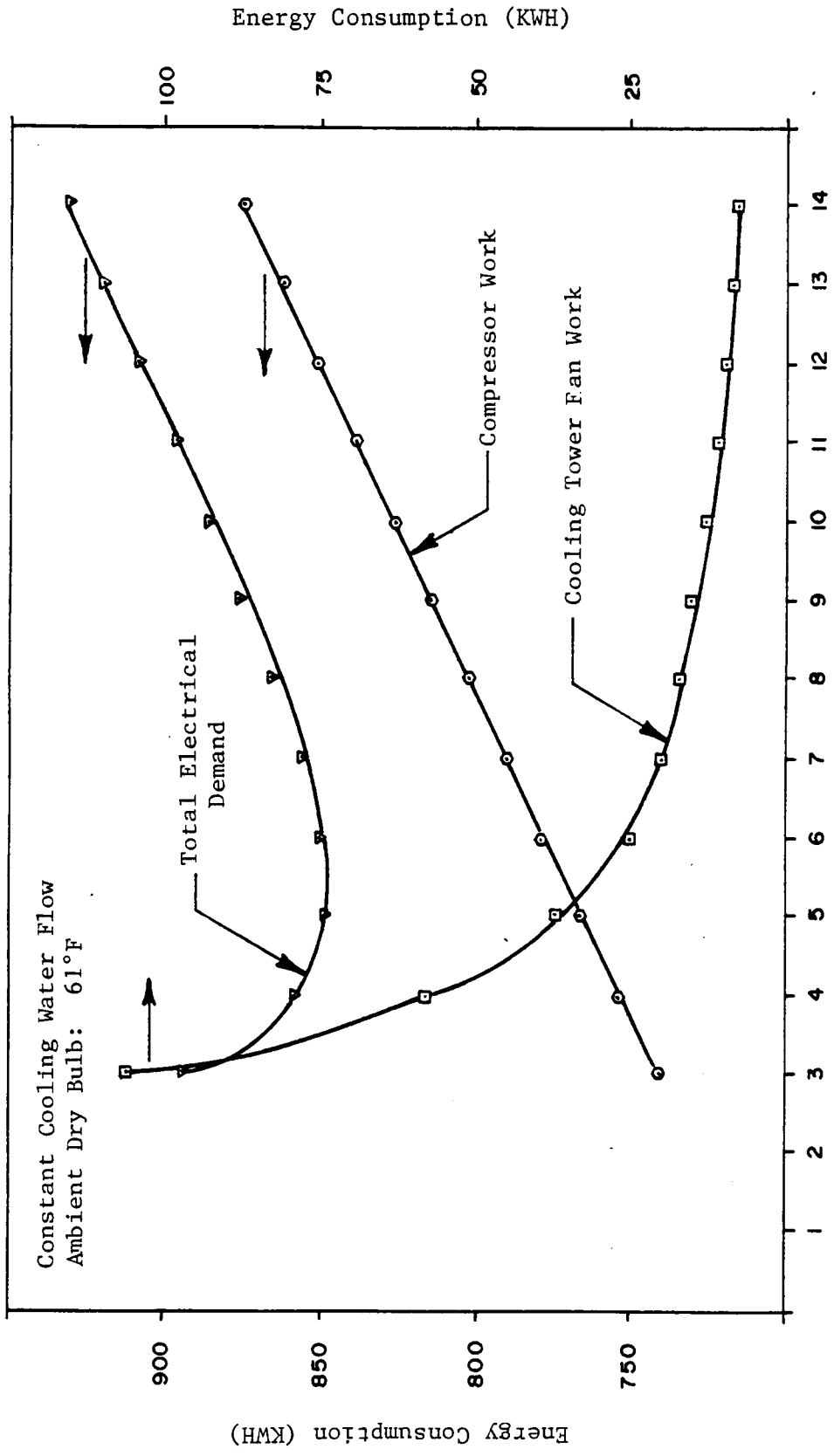


Figure 4.7. Mode 3.2 electrical demand as cooling tower approach temperature varies for 40°F chilled water and an 830 ton load.

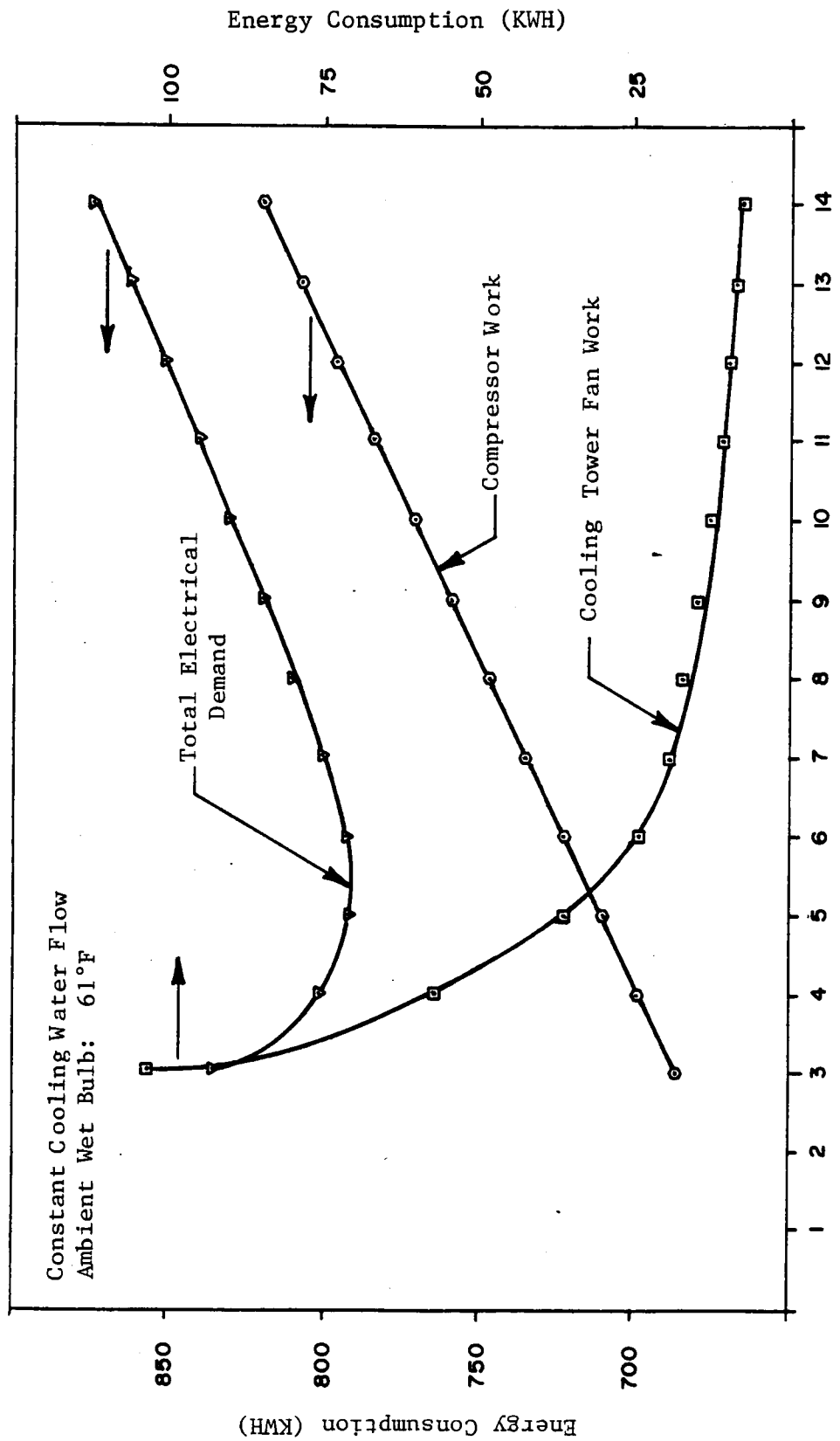


Figure 4.8. Mode 3.2 electrical demand as cooling tower approach temperature varies for 44°F chilled water and an 830 ton load.

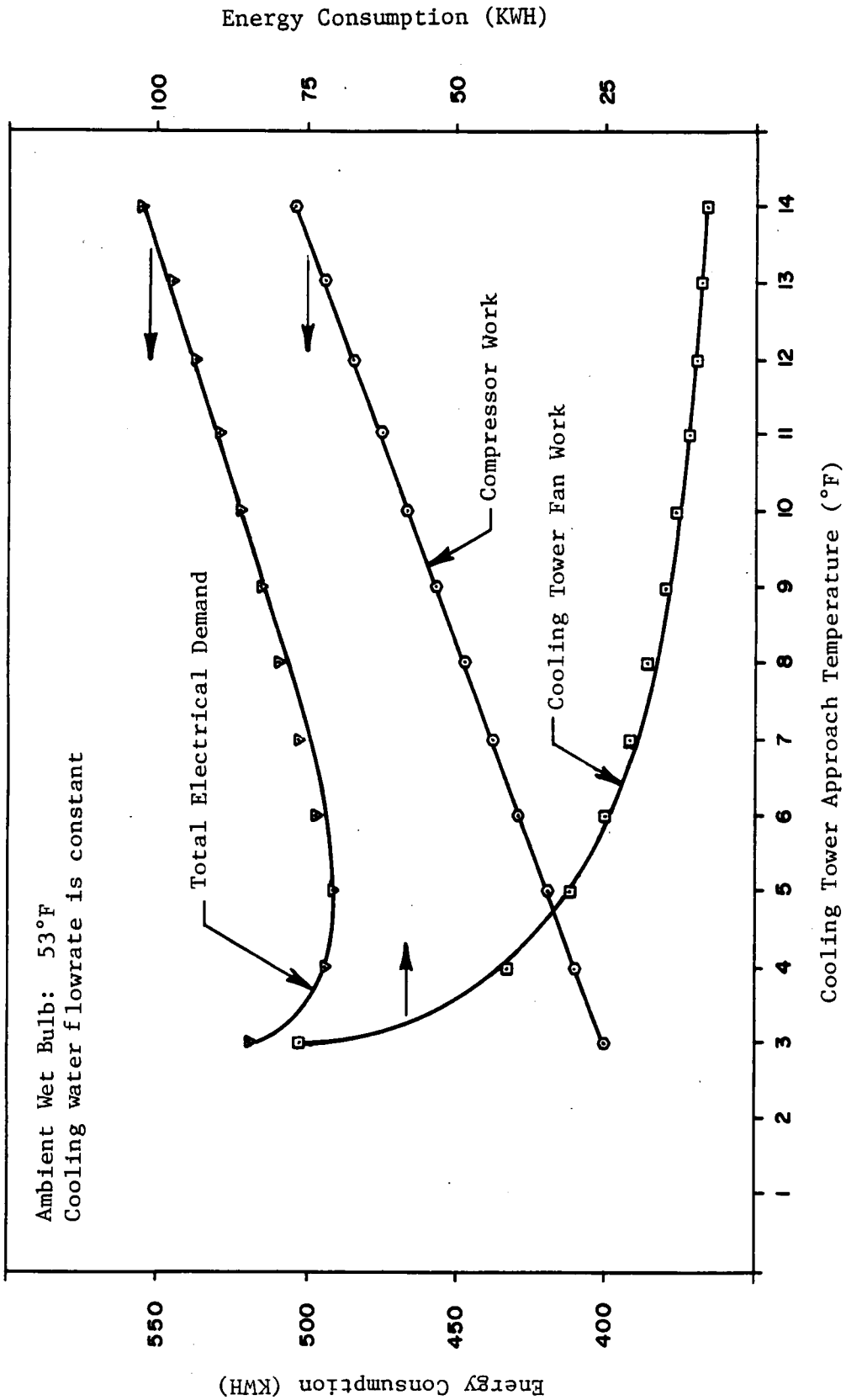


Figure 4.9. Mode 3.2 electrical demand as cooling tower approach temperature varies for 40°F chilled water and a 600 ton load.

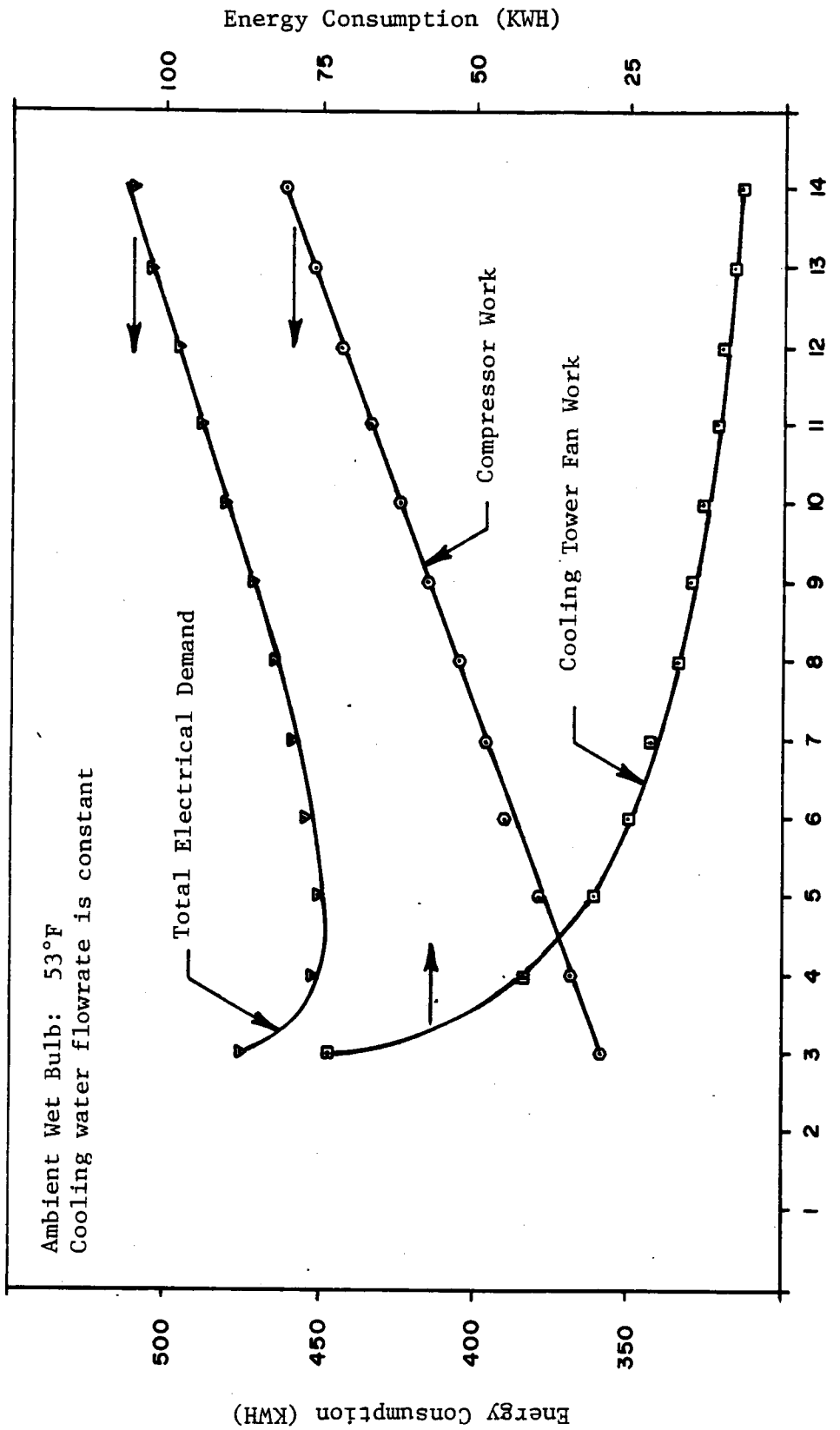


Figure 4.10. Mode 3.2 electrical demand as cooling tower approach temperature varies for 44°F chilled water and a 600 ton load.

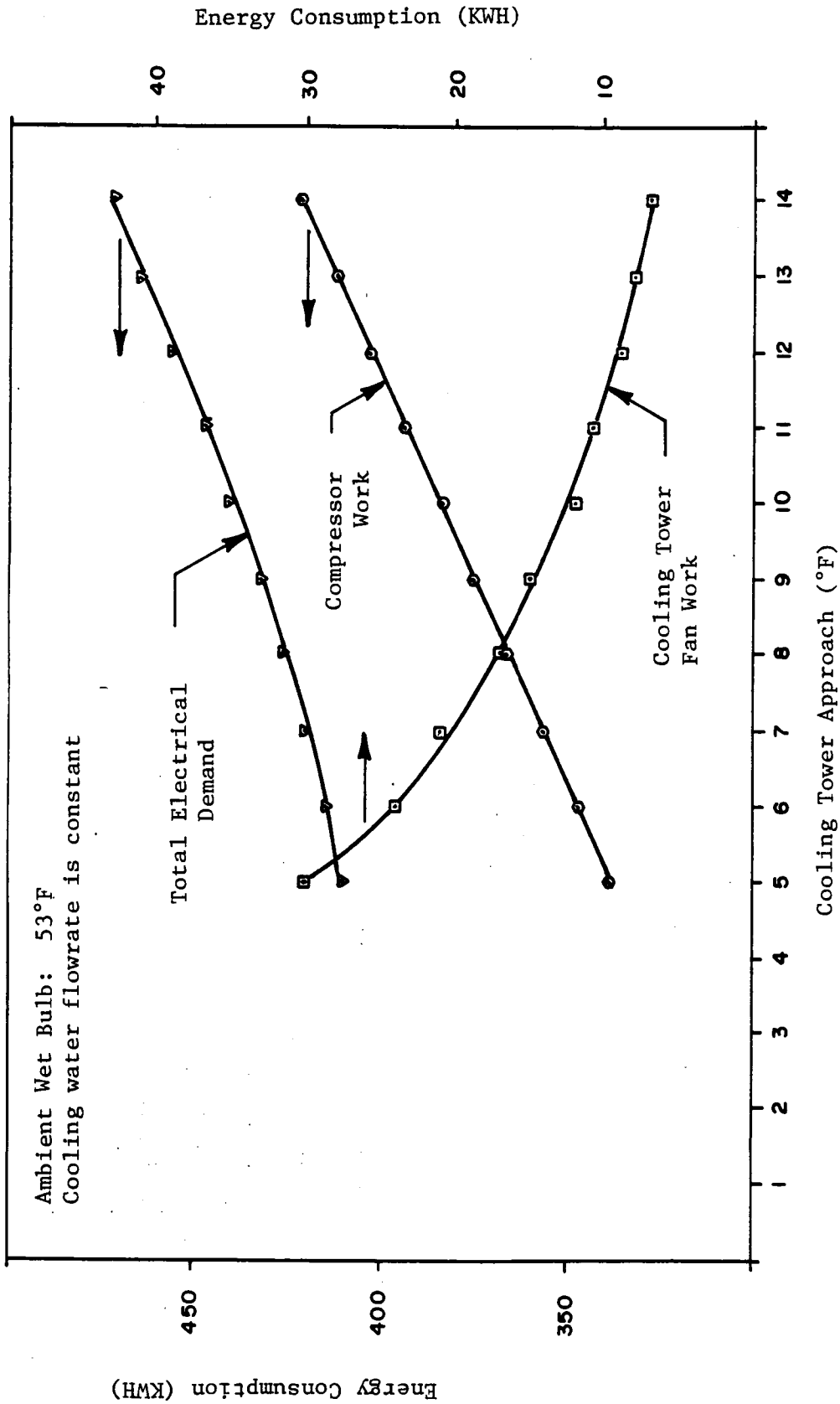


Figure 4.11. Mode 3.2 electrical demand as a function of cooling tower approach for 48°F chilled water and a 600 ton load.

parameter that varies in these three figures is the chilled water supply temperature. It can be seen from the figures that the characteristic curves for the compressor and cooling tower fan work are shifted by changes in temperature but are unaffected in other respects. The compressor work tends to decrease linearly with a decreasing approach about 15 KWH per °F.

The cooling tower fan work appears to be exponentially related to the approach temperature, and the optimum approach temperature occurs where the incremental changes in both curves are equal but of opposite signs. Figures 4.7 and 4.8 present similar studies for an 800 ton load with a 61°F ambient wet bulb. Figures 4.9 to 4.11 conclude the study for a 600 ton load and a 53°F ambient wet bulb. The cooling water range is always in the neighborhood of 10°F for the cooling tower. In all of these studies the shape of the component energy consumption curves are such that a five degree approach temperature is always the appropriate optimum. This result was confirmed often during the course of the study and is presented with confidence in its generality.

As the optimum approach temperature is essentially unaffected by loads, supply temperature, and ambient conditions at constant flow, the next area to explore is the effect of flowrate on the optimum approach.

Figure 4.12 presents a detailed study for an 830 ton load with a 71°F wet bulb. This figure is a contour surface map of the electricity consumption as the approach temperature and cooling water flowrates are varied. This study indicates that optimum cooling water flowrate for these conditions is 2100 gpm and that the 5°F approach is also optimum. The cost function is also strongly unimodal, which means that a straight line from any point on the surface to the optimum is strictly decreasing.

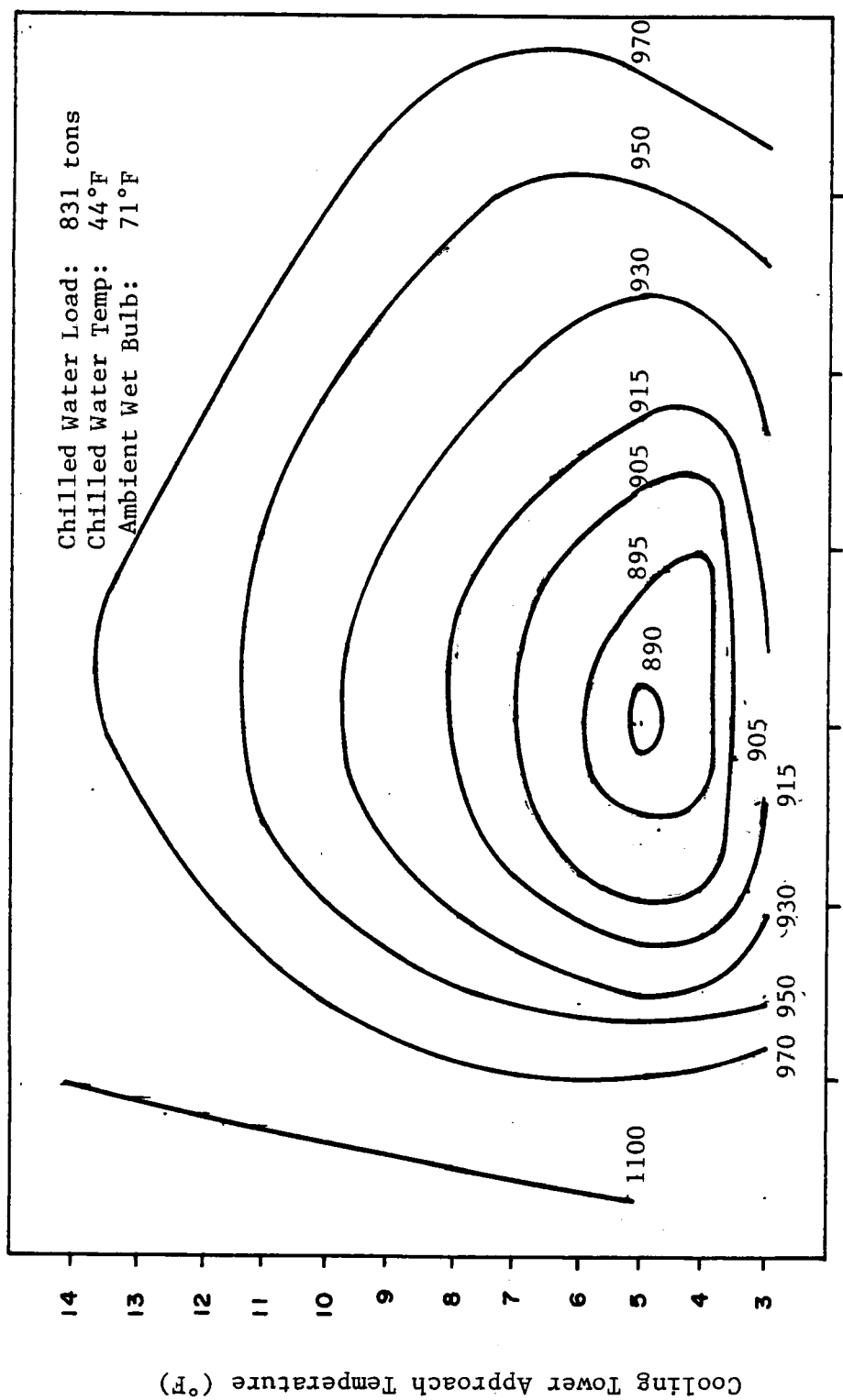


Figure 4.12. Constant cost contours as both cooling tower approach temperature and cooling water flow vary for mode 3.2

In the neighborhood of the optimum approach the cost surface is also relatively flat. A 500 gpm change in flow will only increase the energy consumption by 1.5 to 3 percent, which translates into $.70/hour for electricity at $0.045/KWH. Considering that the analog control systems cannot be expected to control flow much more accurately than this, a flow set point which is within 500 gpm of the optimum is adequate.

Hereafter, the five degree approach temperature set point will be considered the optimum for all sensitivity studies in Mode 3.2. Later in the discussion of Mode 3.4 (1250 ton steam chiller) it will be shown to be general.

Constant cooling water flowrate. The next area to explore is the cooling water flowrate optimization. Figures 4.13 to 4.16 illustrate a typical set of studies which were performed to verify that the optimum cooling water flowrate can be represented by a constant flow of cooling water. These studies are similar to the initial studies used to justify the approach temperature optimization's insensitivity to load, supply temperature and wet bulb. Figures 4.13 and 4.14 show that regardless of the load or ambient wet bulb, the optimum flowrate is unaffected by the chilled water supply temperature. Figure 4.15 shows that for a fixed load and supply temperature the ambient wet bulb does not affect the optimum flowrate. Figure 4.16 illustrates the effect of load on the optimum flowrate for one set of conditions. Since in Figures 4.13 to 4.15 it has been shown that the optimum is insensitive to the values of these conditions, the optimization results of Figure 4.16 are general over the range studied. (The data used to construct Figure 4.16 is provided on Table 4.1.) In the figure are plotted two dashed lines. One of these

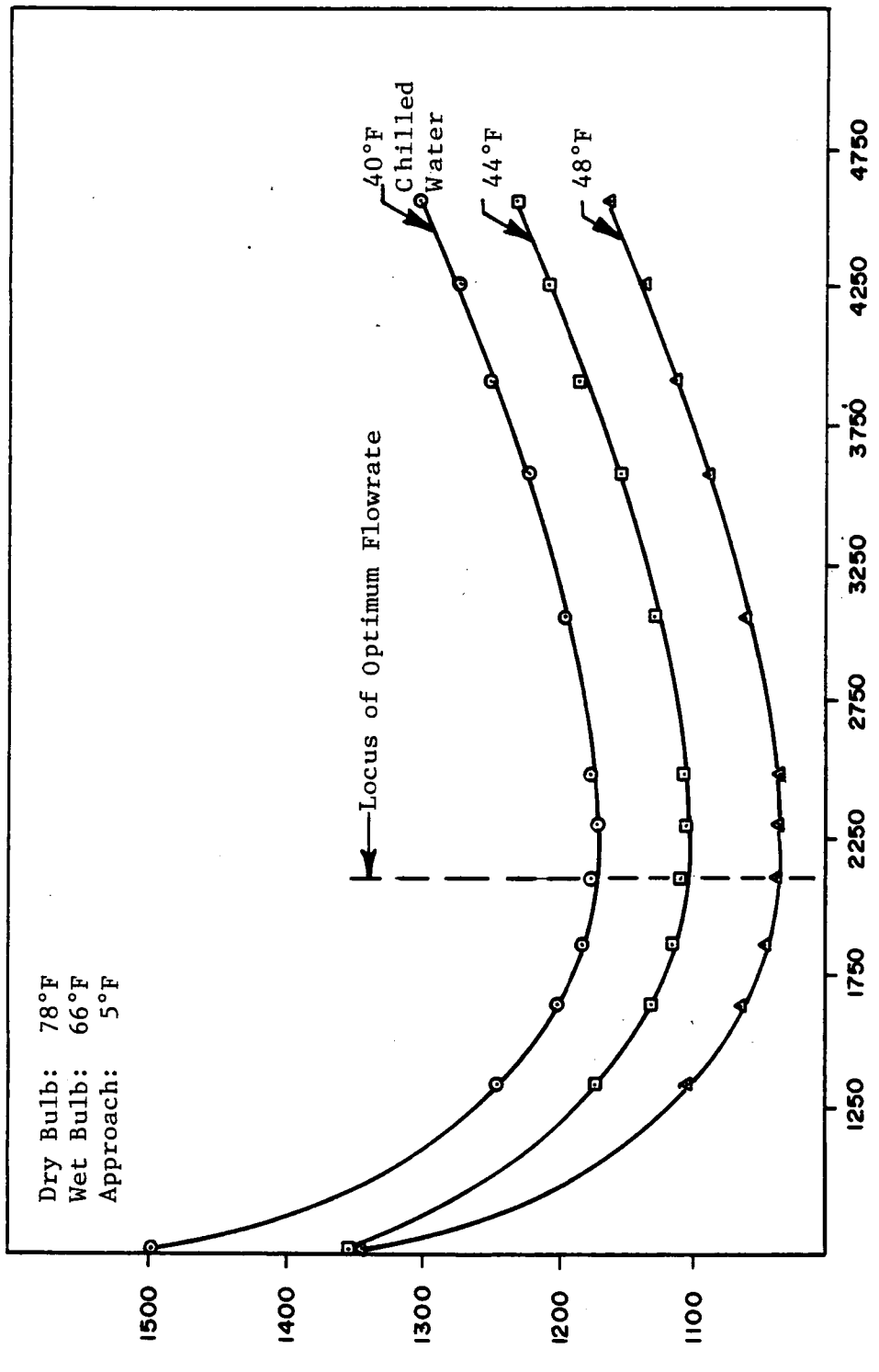


Figure 4.13. Effect of chilled water supply temperature on the optimum cooling water flowrate of Mode 3.2 for a 930 ton load.

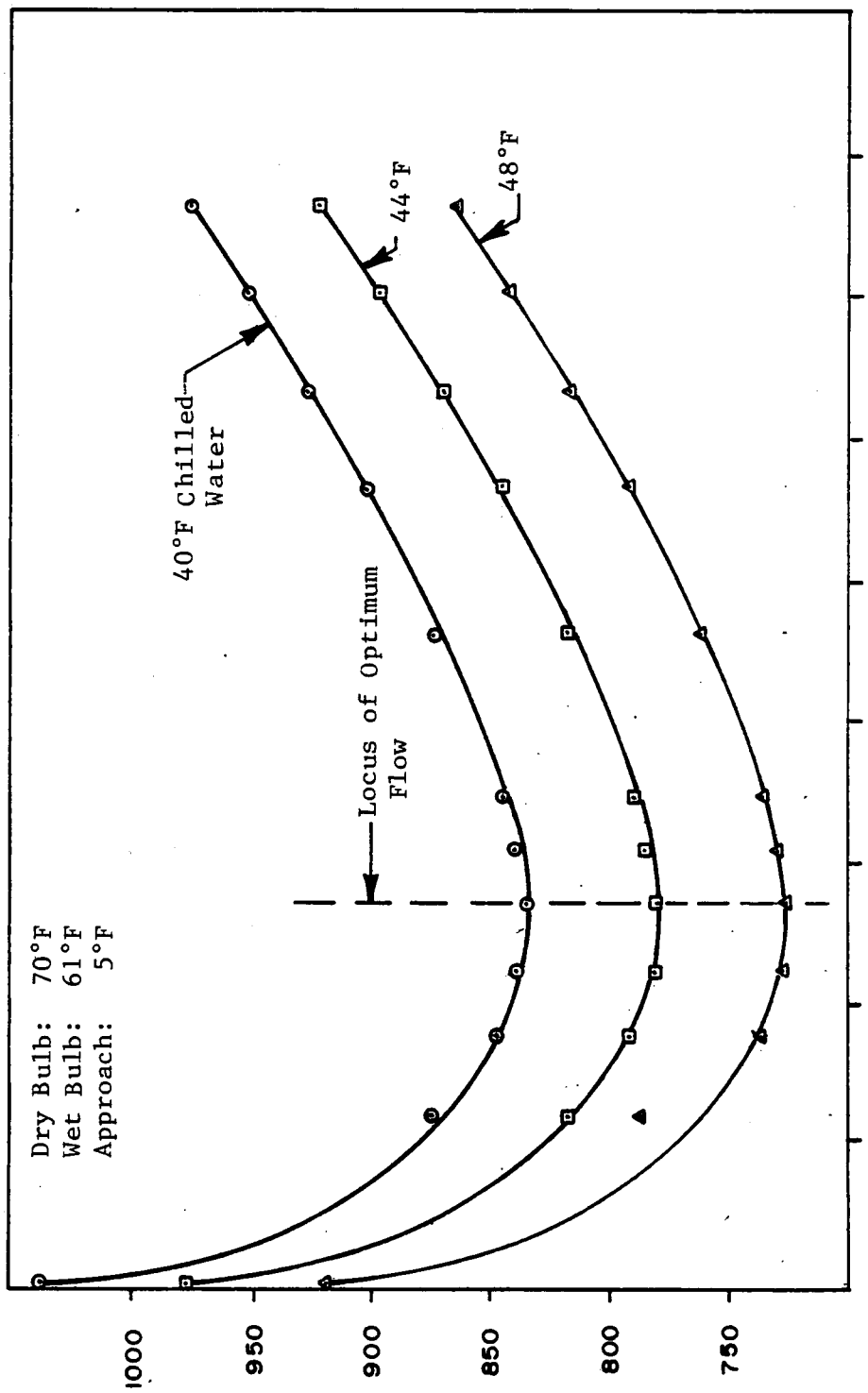


Figure 4.14. Effect of chilled water supply temperature on the optimum cooling water flowrate of mode 3.2 for an 830 ton load.

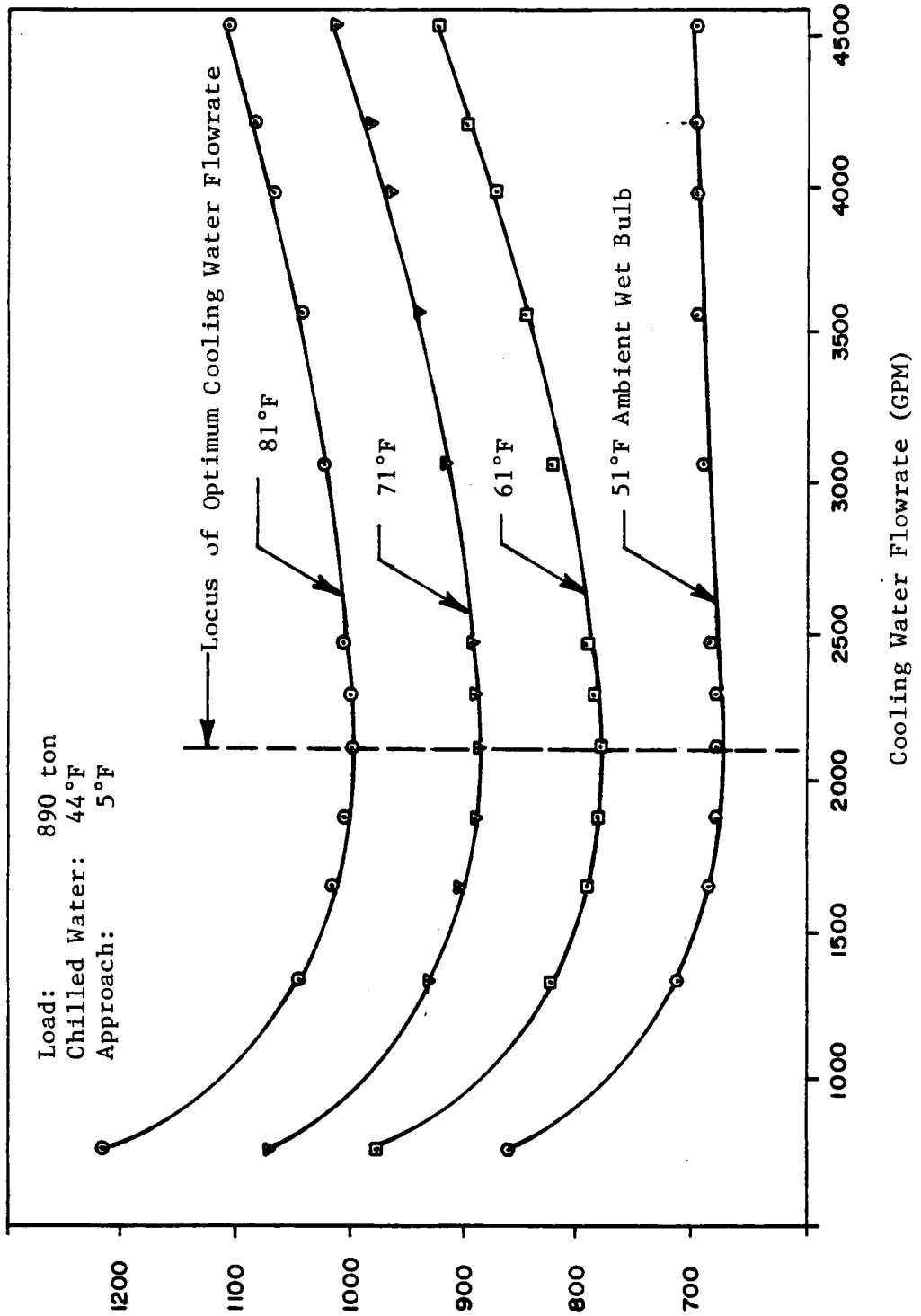


Figure 4.15. Effect of ambient wet bulb temperature on the optimum cooling water flowrate for Mode 3.2.

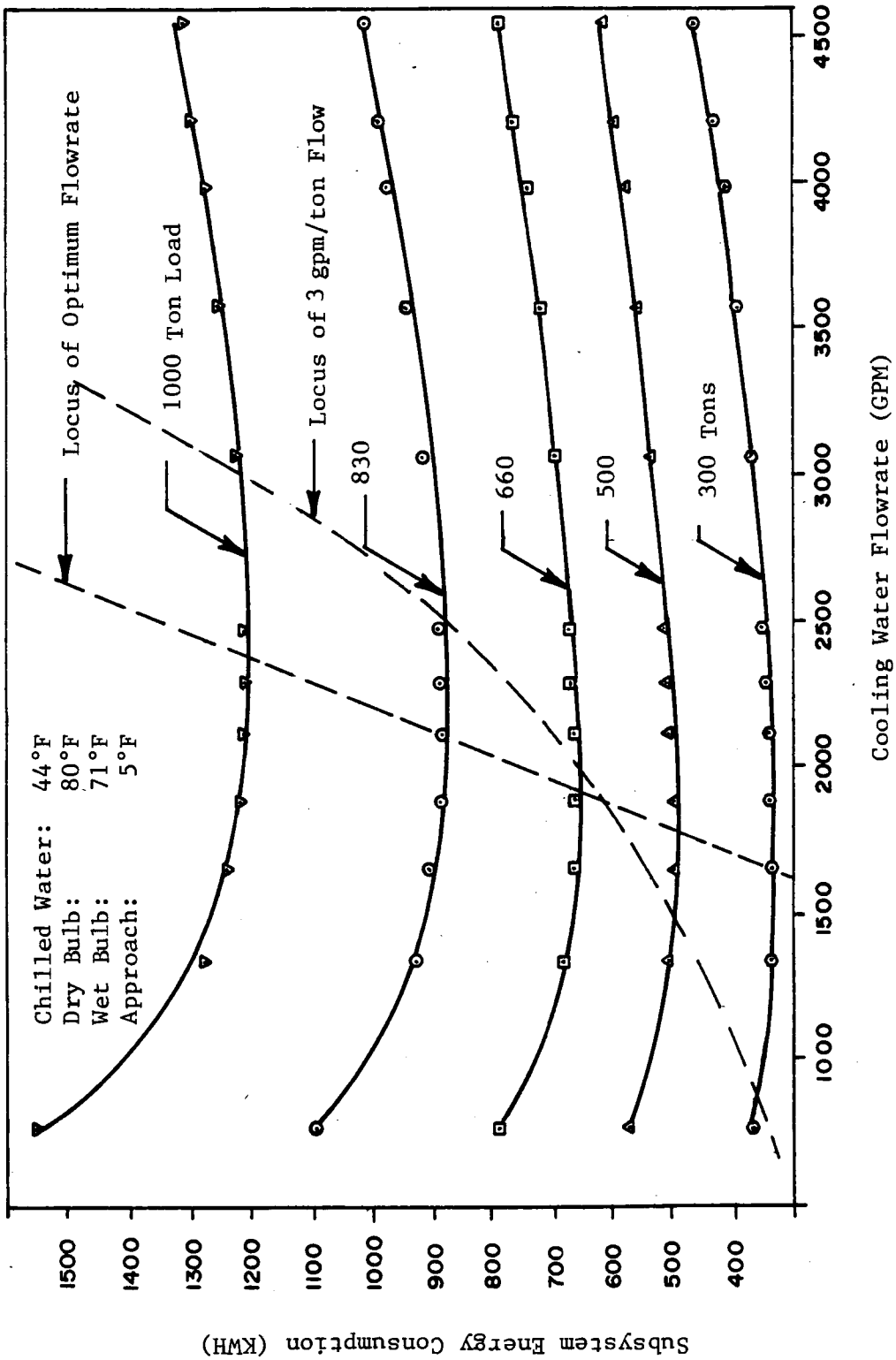


Figure 4.16. Effect of subsystem load on the optimum cooling water flowrate for Mode 3.2 (a 1250 ton electric chiller).

TABLE 4.1

ENERGY COST OF MODE 3.2 AS A FUNCTION OF THE COOLING WATER FLOWRATE*

Flowrate (GPM)	Load (tons) 1000	831	660	500	300
			Cost (KWH)		
760	1551	1097	791	567	361
1330	1285	930	685	504	336
1636	1240	903	670	499	336
1885	1220	892	667	499	339
2103	1212	889	669	502	343
2297	1201	891	672	507	348
2474	1209	895	677	512	353
3060	1229	918	700	534	373
3576	1254	943	723	555	394
3916	1279	966	745	576	414
4255	1303	989	767	597	434
4556	1326	1012	789	618	455

*Chilled Water Temperature: 44°F
Ambient Dry Bulb: 80°F
Ambient Wet Bulb: 71°F
Cooling Tower Approach: 5°F

lines is the locus of constant 3 gpm/ton load, the other is the locus of the optimum flowrate. The 3 gpm/ton curve represents the optimum flowrate suggested by ASHRAE, and the other is the true optimum. In the previous discussion on the sensitivity of the subsystem electrical consumption to cooling water flowrate, it was shown that very little penalty is incurred if flow is within 500 gpm of the optimum. The locus of the optimum flowrate always lies within 500 gpm of a 2000 gpm flowrate, whereas the constant 3 gpm/ton curve deviates considerably from the optimum at high and low loads.

It is concluded that the optimization algorithm for a single electric chiller is quite simple. The control systems should maintain a constant cooling tower approach and a constant cooling water flowrate. The simulation indicates the values to be about 5°F and 2100 gpm. In actual building operation the values may be somewhat different but since the studies show them to be independent and constant, the actual values could be determined experimentally quite easily.

Steam Chiller Mode 3.4

Earlier in this report it was stated that if the optimization results for a single electric chiller were true for a single steam chiller then it would be assumed true for all of the chiller combinations to be studied. This conclusion is based on the assumption that two chillers in parallel behave as one large chiller. A recent paper by Newton and Beekman (20) in the ASHRAE Transactions agrees with this conclusion.

Figures 4.17 to 4.20 show the results of a study of Mode 3.4. Energy Costs are in terms of equivalent kilowatt hours. The steam consumption was converted to kilowatts by multiplying the ratio of steam cost

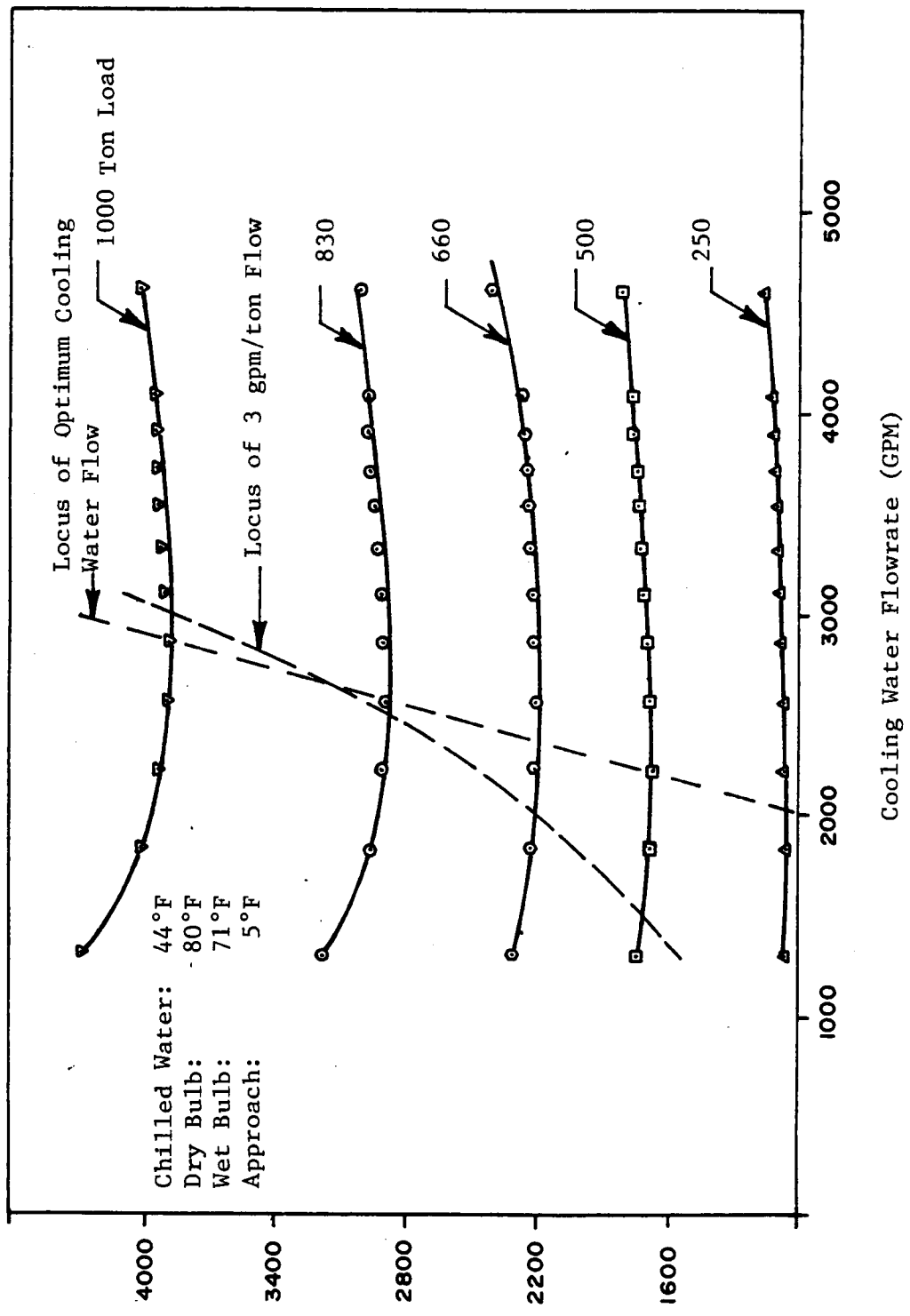


Figure 4.17. Effect of subsystem load factor on the optimum cooling water flowrate for Mode 3.4 (1250 ton steam chiller).

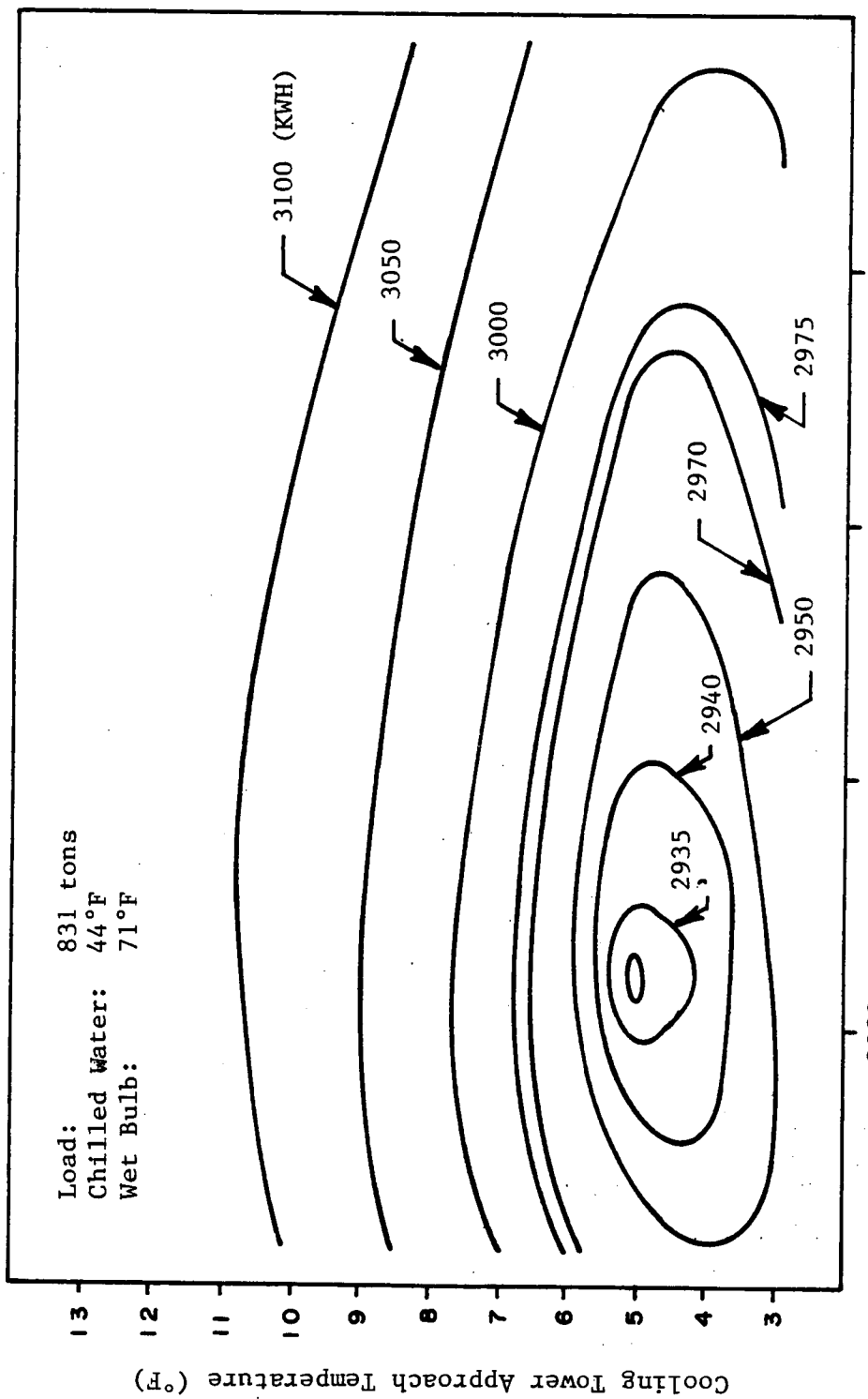


Figure 4.18. Constant cost contours as both cooling tower approach temperature and cooling water flowrate are varied for Mode 3.4.

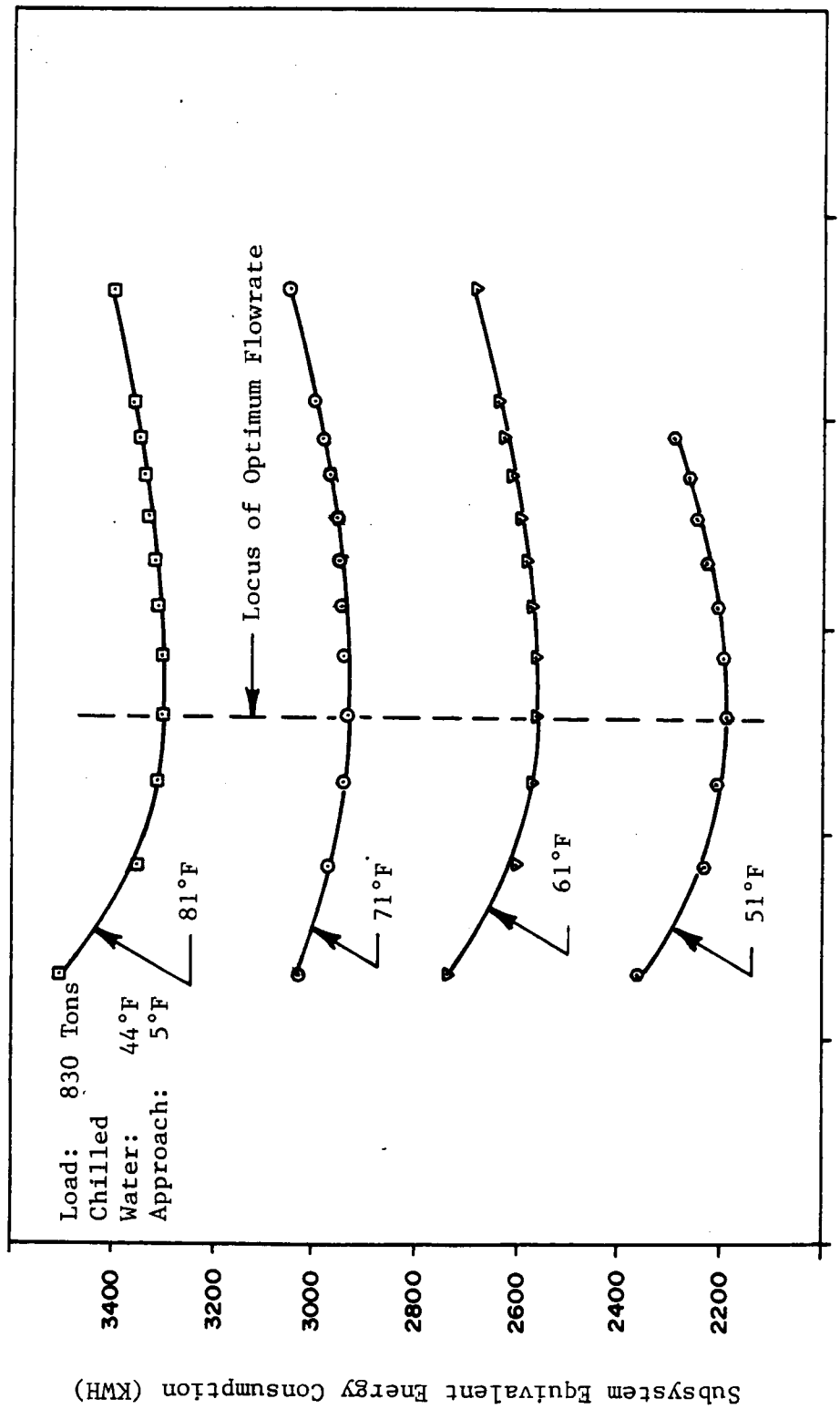


Figure 4.19. Effect of ambient wet bulb temperature on the optimum cooling water flowrate of Mode 3.4.

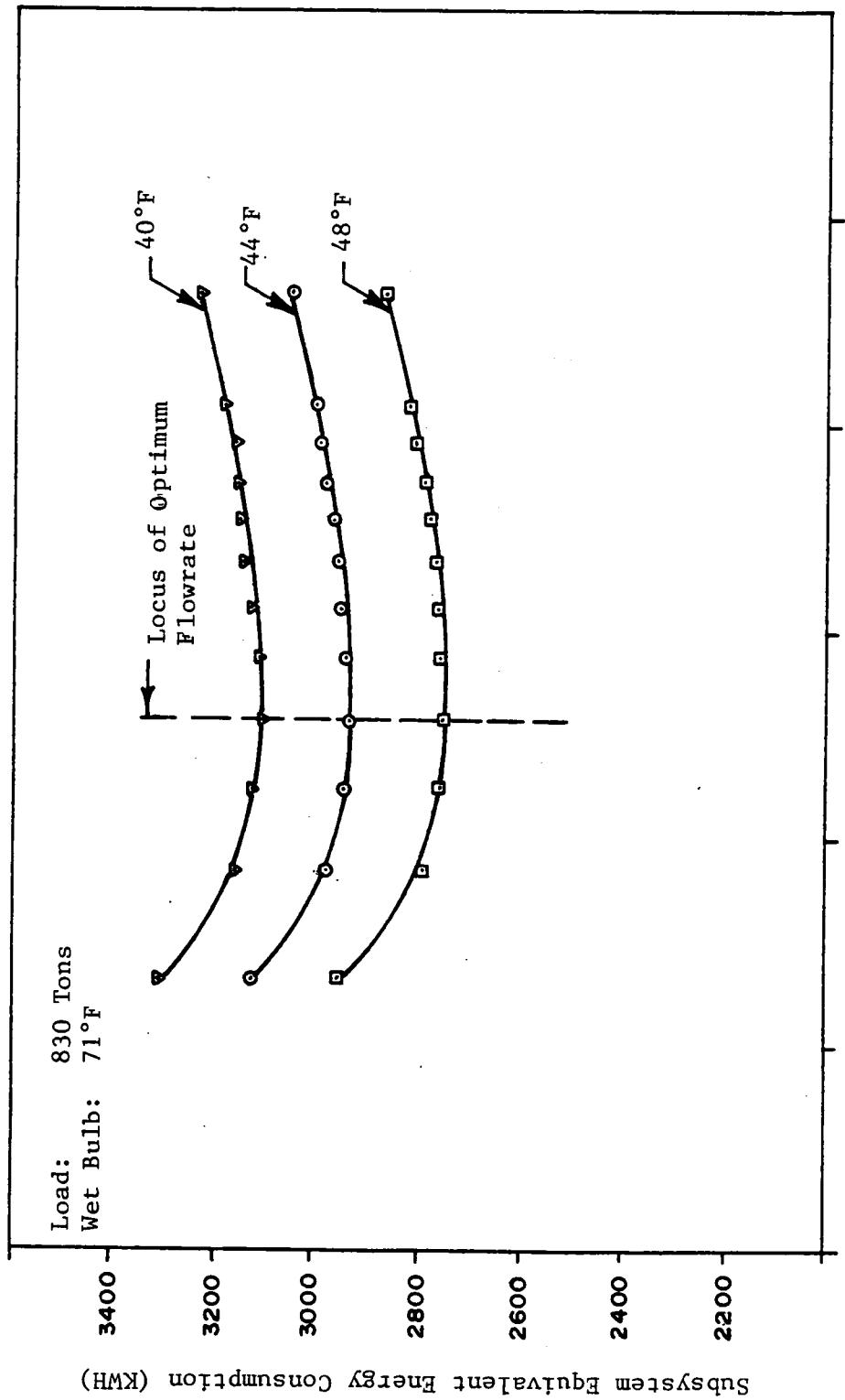


Figure 4.20. Effect of chilled water supply temperature on the optimum cooling water flowrate of Mode 3.4.

to electricity cost. The unit cost used for steam was $9/1000 lbs. This figure was supplied by IBM and was presumed to be current for mid-1979. From these studies it is concluded that the cooling tower optimization is not changed for Mode 3.4. The value of the cooling water flowrate is greater (see Table 4.2), reflecting the difference in the chiller's character, but the optimum approach is still 5 degrees. The optimization algorithm need only provide a new flowrate to the first level controllers to update the optimization.

Optimization of the Operation of Two Parallel Chillers Mode 3.3

It was explained in the previous section that the operation of two chillers in parallel may be simplified for the optimization of the cooling water flow loop. As has been described previously the chilled water pumping network optimization completely determines the condition of that loop. The only optimization left for the operation of two parallel chillers is the distribution of the load between chillers. In this optimization there are two handles; one handle involves operating the chillers at different supply temperatures and mixing to achieve the required chilled water. The other handle involves how the total flow is distributed between chillers. This discussion will show that there is an easily implemented procedural optimization which will minimize the utility consumption of two parallel chillers.

Figure 4.21 presents a schematic diagram for Mode 3.3. In this submode both electric chillers are running. The figure presents a control scheme which is illustrative in nature and is not intended to be comprehensive. The purpose of the figure is to show how the procedural and feedback optimizations interact.

TABLE 4.2

ENERGY COST OF MODE 3.4 AS A FUNCTION
OF THE COOLING WATER FLOWRATE*

	Load (tons)				
	1000	830	660	500	250
Flowrate (GPM)			Cost (KWH)		
1305	4235	3120	2314	1750	1069
1840	4009	2977	2225	1697	1061
2242	3953	2942	2206	1961	1067
2573	3938	2936	2205	1698	1076
2857	3937	2938	2212	1709	1087
3110	3941	2943	2223	1721	1098
3336	3947	2950	2236	1732	1108
3543	3954	2958	2248	1744	1118
3734	3962	2971	2260	1756	1129
3911	3969	2983	2272	1767	1140
4076	3977	2996	2284	1778	1150
4646	4013	3044	2430	1822	1191

*Chilled Water Temperature: 44°F
Ambient Dry Bulb: 80°F
Ambient Wet Bulb: 71°F
Cooling Tower Approach: 5°F

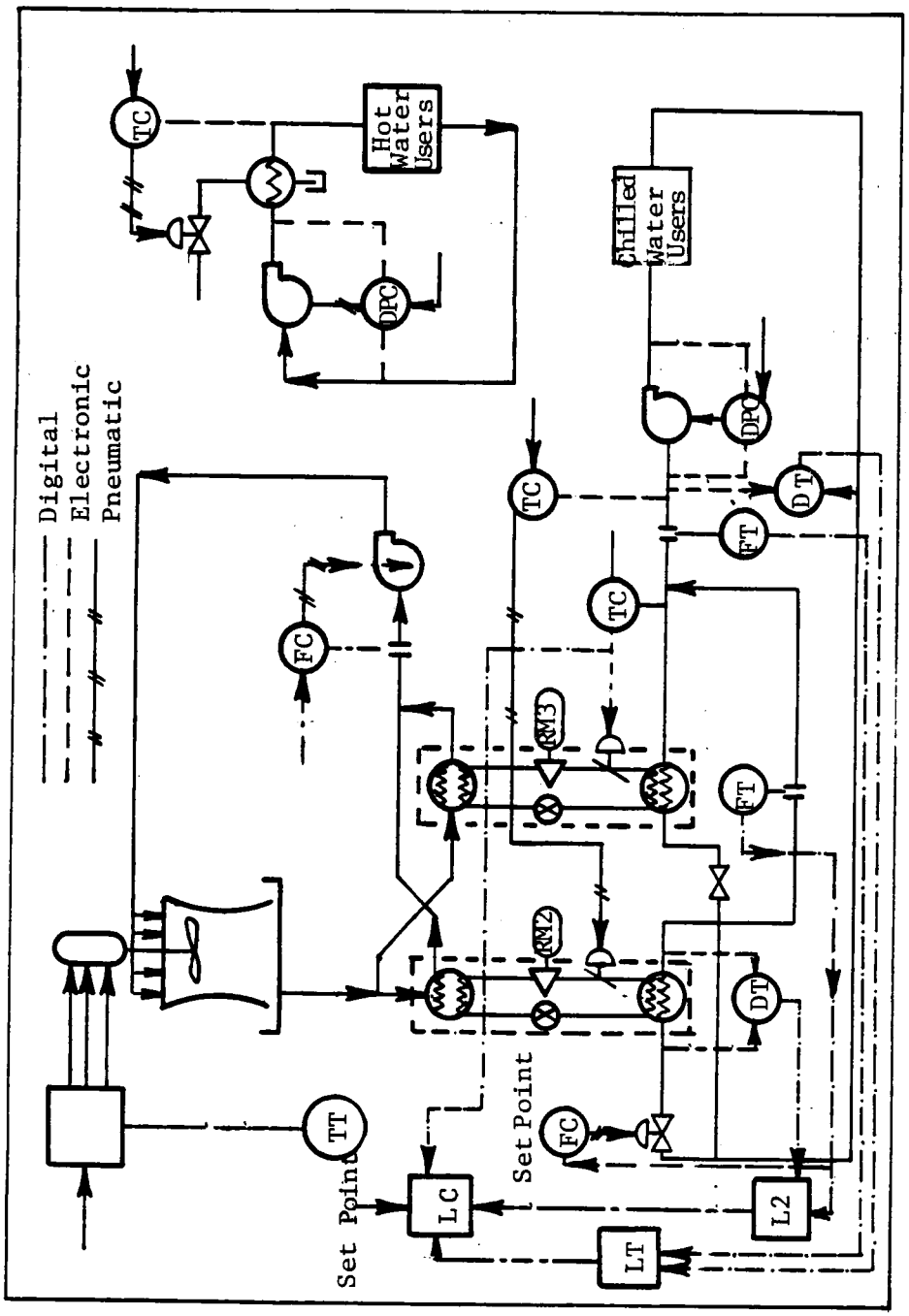


Figure 4.21. Mechanical refrigeration Mode 3.3.

The cooling tower/cooling water loop is unchanged. Figure 4.21 shows two rectangular boxes labeled L2 and LT. These represent calculation blocks which compute the total chilled water load (LT) and the load on the large chiller (L2). The results from these calculations are then given to a load controller (LC) which provides a set point to flow controllers which distributes the total flow between the chillers. The total mixed chilled water temperature is controlled by adjusting the large chiller's vanes. The small chiller chills its own water to a temperature requested by the optimization algorithm. The output of the small chiller temperature controller provides a limit to the load controller. This limit prevents the load controller from overloading the small chiller. It is assumed that the flow ratio controllers always maintain one of the chiller flow valves 100% open. This control scheme will allow the optimization algorithm to specify the load distribution between the two chillers and the desired small chiller temperature set point. This automatically establishes the flow ratio between the chillers for a given load.

A set of studies were made to determine the optimum load distribution between chillers and the optimum set point to the small chillers for three loads. The results of these studies are presented in Figures 4.22 to 4.24. The three figures show that the optimum load distribution occurs around 70%, unless the small chiller's compressor is constrained. The optimum set point to the small chiller is around 2 degrees below the mixed chilled water temperature for a medium load, but total subsystem cost is very insensitive to this parameter. Savings for properly distributing the load between chillers ranged from 2 to 3 percent.

Figures 4.25 and 4.26 explain why the optimum load distribution occurs around 70%. This figure is for a 750 ton load (about 50% of

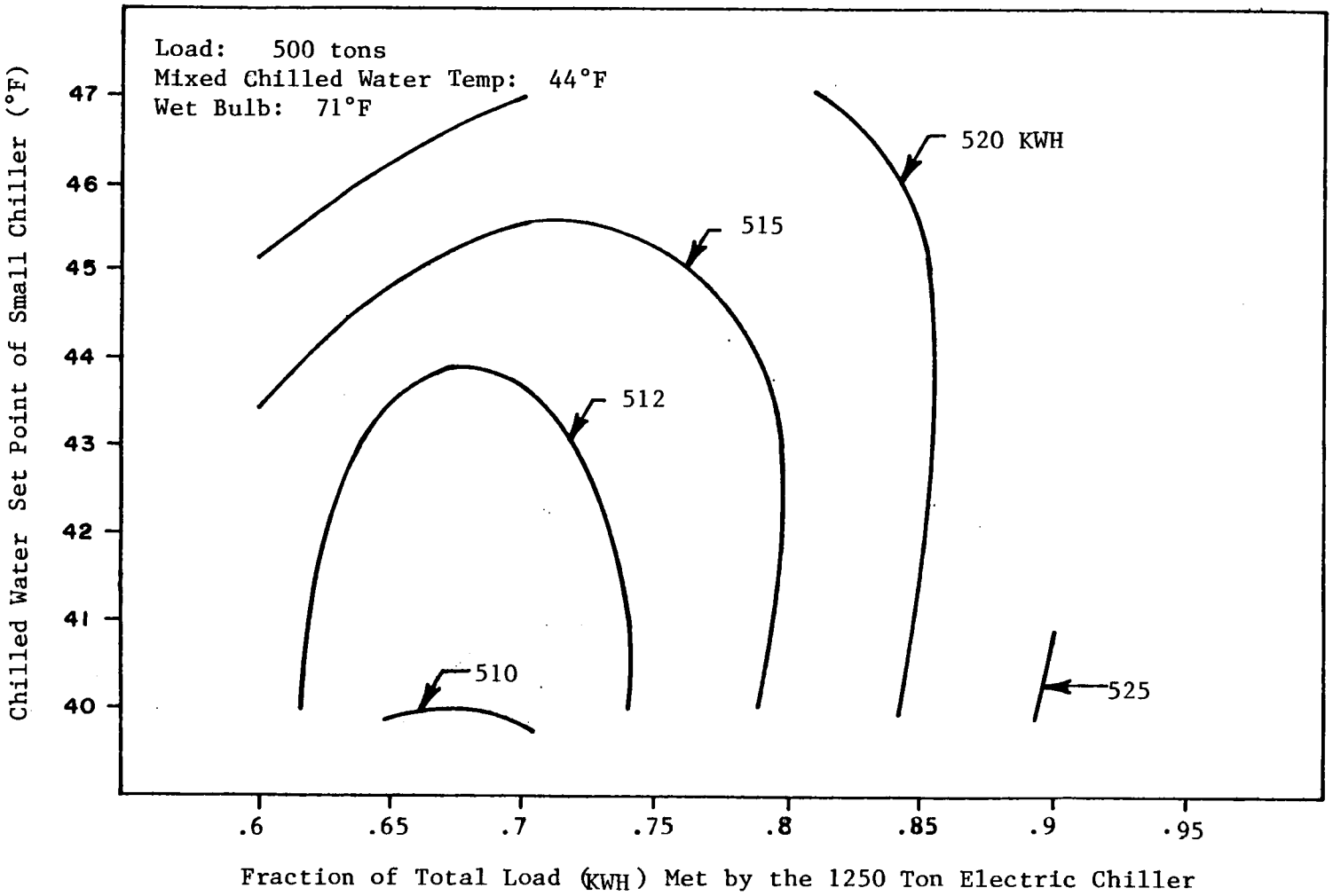


Figure 4.22. Constant cost contours for Mode 3.3 as both the load distribution between chillers and the small chiller supply temperature are varied for a 500 ton load.

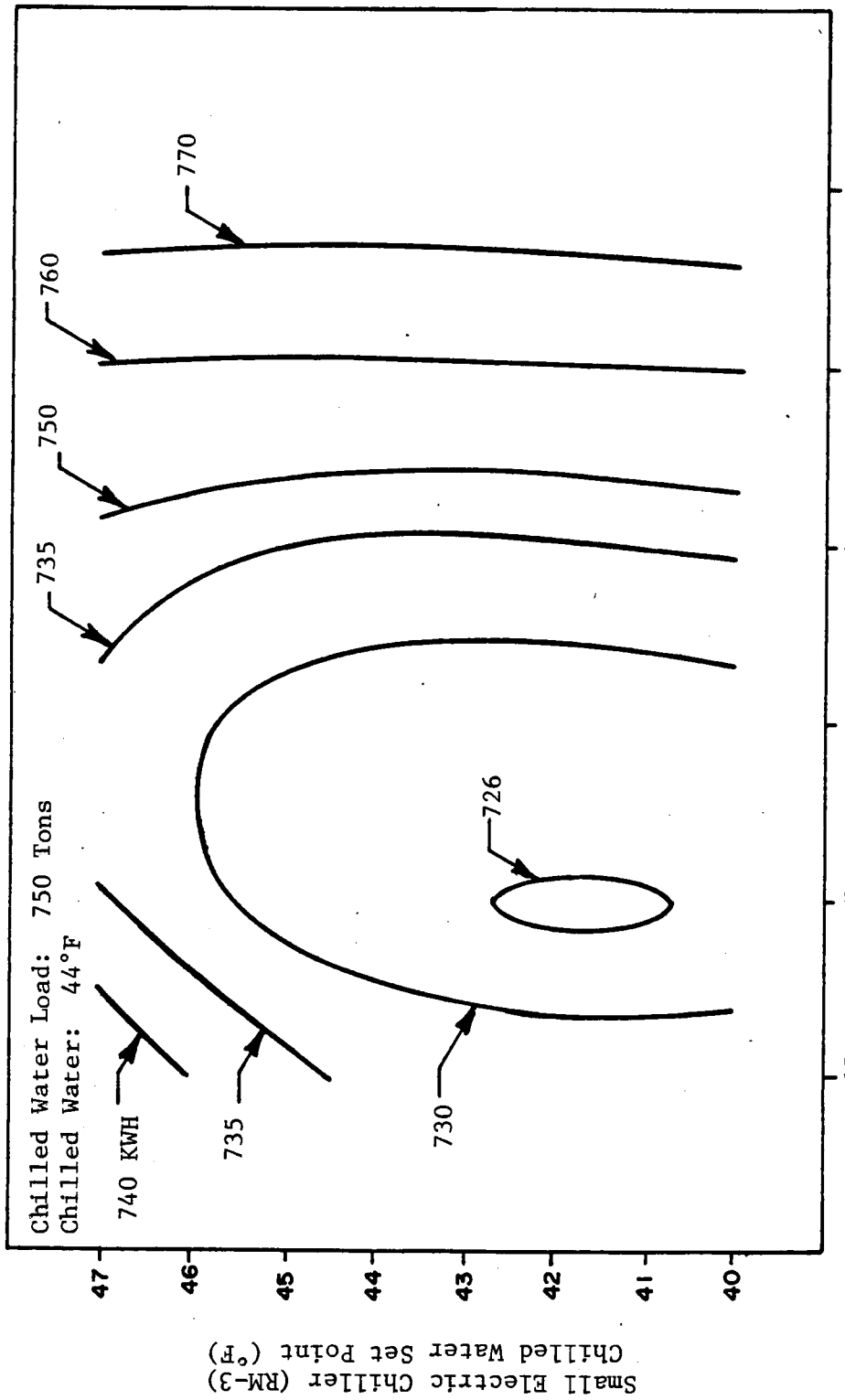


Figure 4.23. Contour surface map of subsystem Mode 3.3 power consumption.

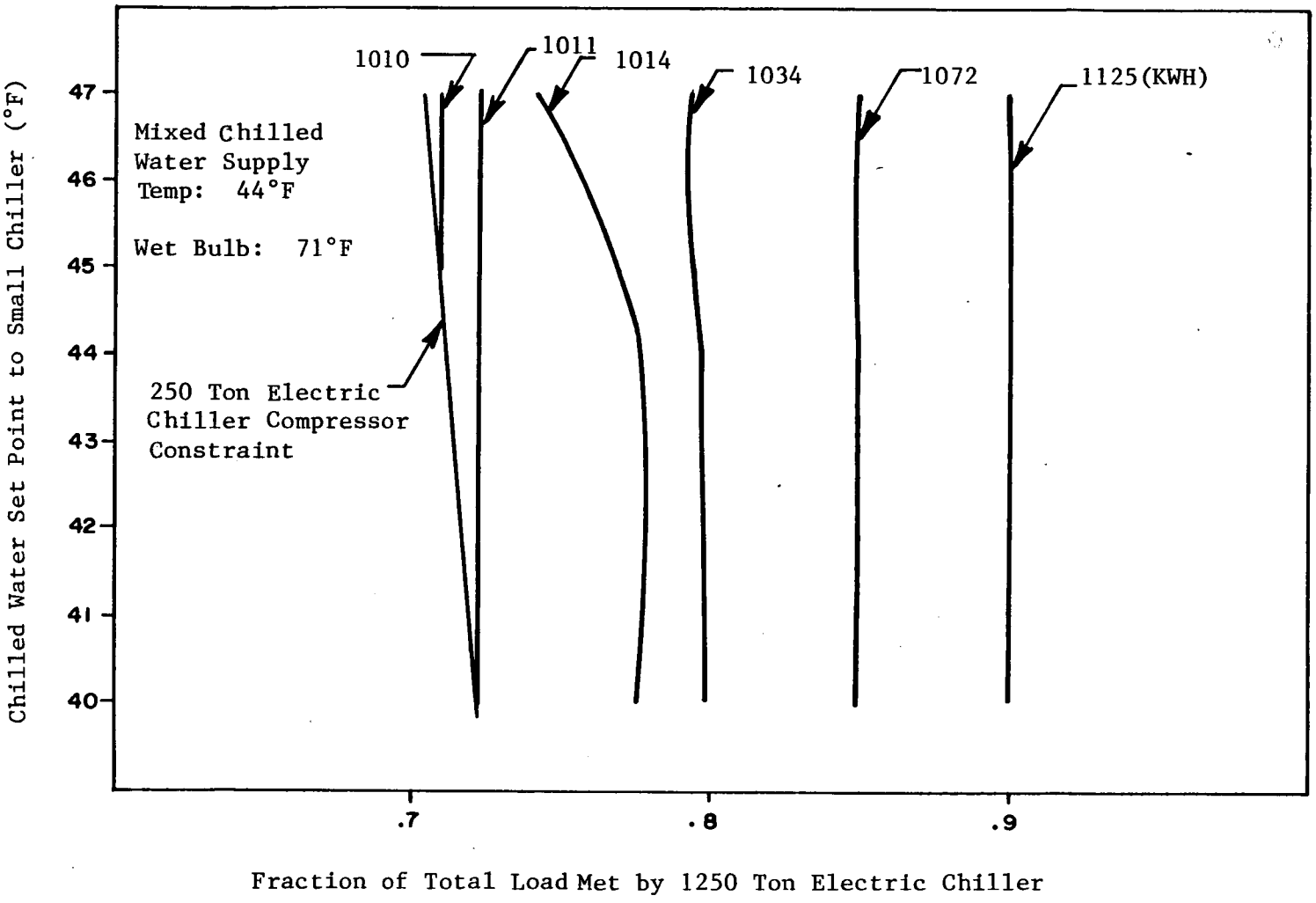


Figure 4.24. Constant cost contours for Mode 3.3 as both the load distribution and small chiller supply temperature are varied for a 1000 ton load.

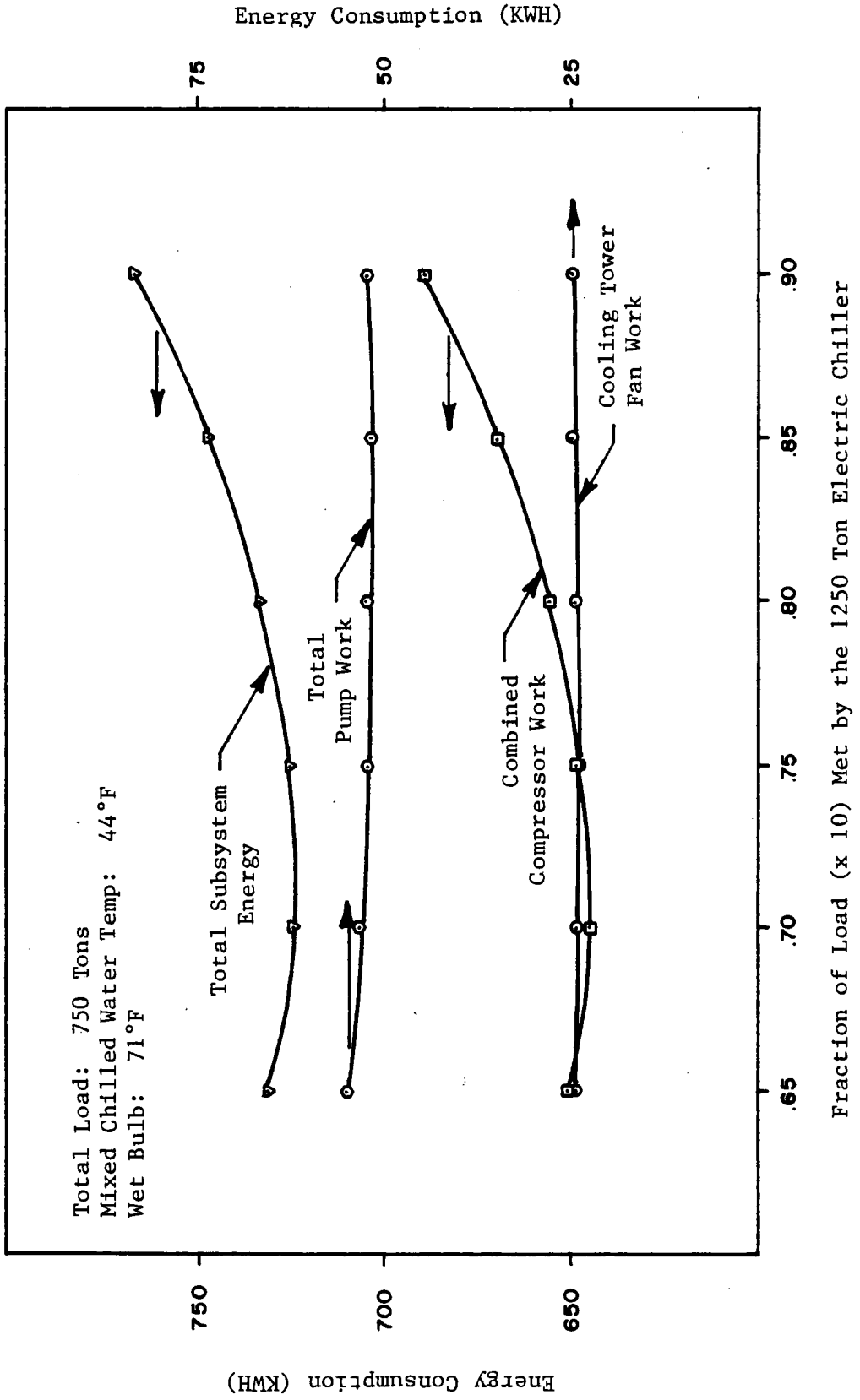


Figure 4.25. Contribution of the various components in Mode 3.3 to the total subsystem energy consumption as the load distribution between the chillers is varied. (Small chiller set point is 42°F.)

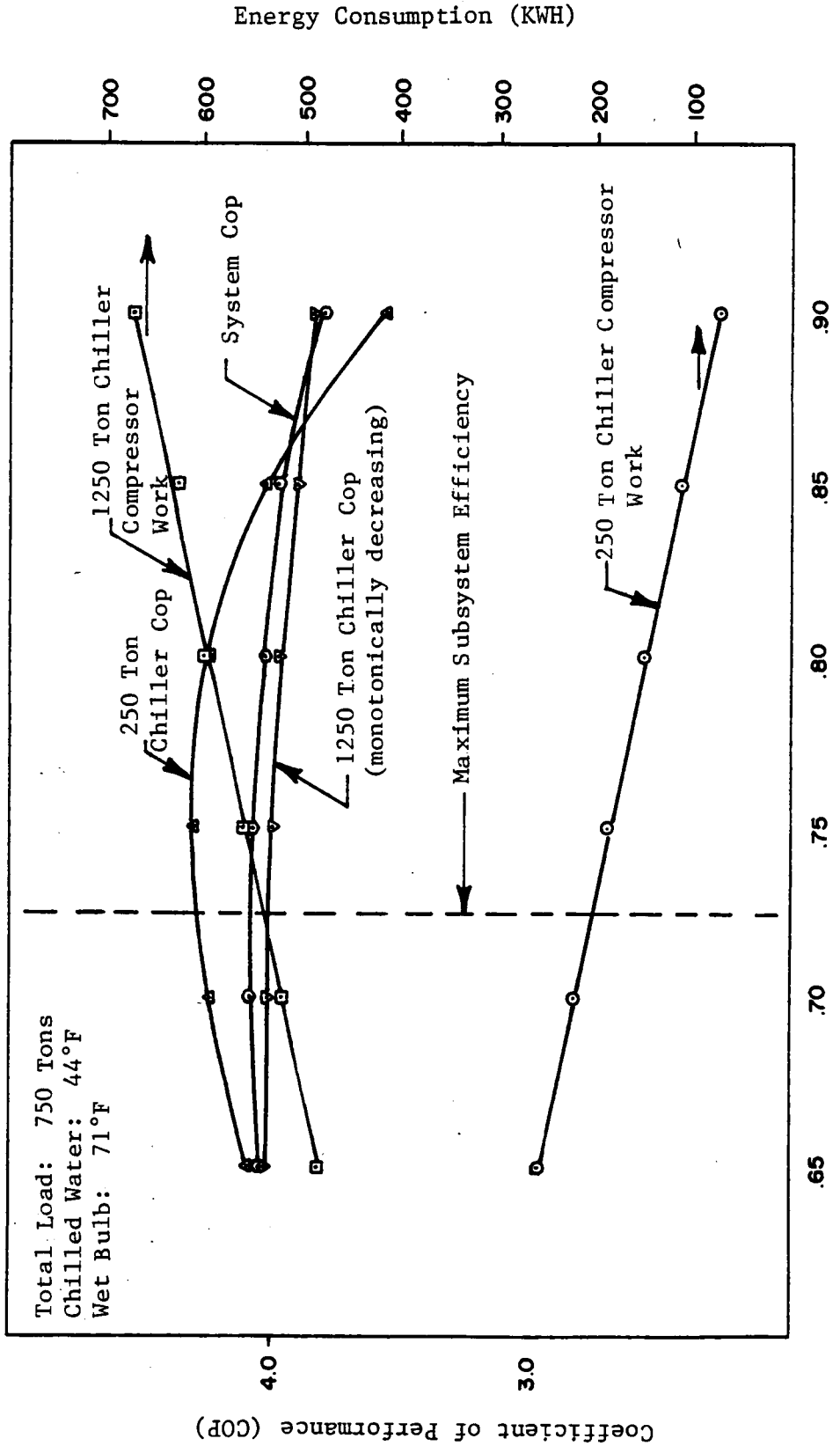


Figure 4.26. Various factors which contribute to the location of the optimum load distribution between the two electric chillers for Mode 3.2.

capacity) with the small chiller providing 2 degrees colder chilled water. As the small chiller is unloaded its efficiency increases until it is operating at 80% of its capacity and then it decreases. The efficiency of the large chiller is always decreasing under these conditions. The system efficiency shows a peak around 72%. This is the point where the incremental increase in RM-3's efficiency is just balanced by the incremental decrease in RM-2's efficiency. Figure 4.25 shows that the combined compressor cost determines the location of the optimum distribution.

The optimization algorithm for Mode 3.3 is quite simple. From Figure 4.27 it can be seen that a constant cooling water flowrate of 2500 gpm is appropriate. Table 4.3 contains the numerical data used to construct Figure 4.27. The cooling water approach temperature should be 5°F. The load should be distributed so that the small chiller meets 30% of the total load. If the small chiller cannot meet 30% of the total load, it should meet the maximum possible percent. The small chiller may be operated with its supply temperature equal to the total chilled water supply temperature, but for light to medium loads 2 degrees colder is better.

Parallel Steam and Electric Chiller Mode 3.5

Mode 3.5 is another important submode. This configuration has the 1250 ton steam chiller and the 250 ton electric chiller running. In Figure 4.28 a schematic of the equipment configuration and a simple optimization scheme is presented. The control strategy for Mode 3.5 differs from the strategy of Mode 3.3 in that the small chiller is always base loaded. In consideration of the optimization's sensitivity to utility

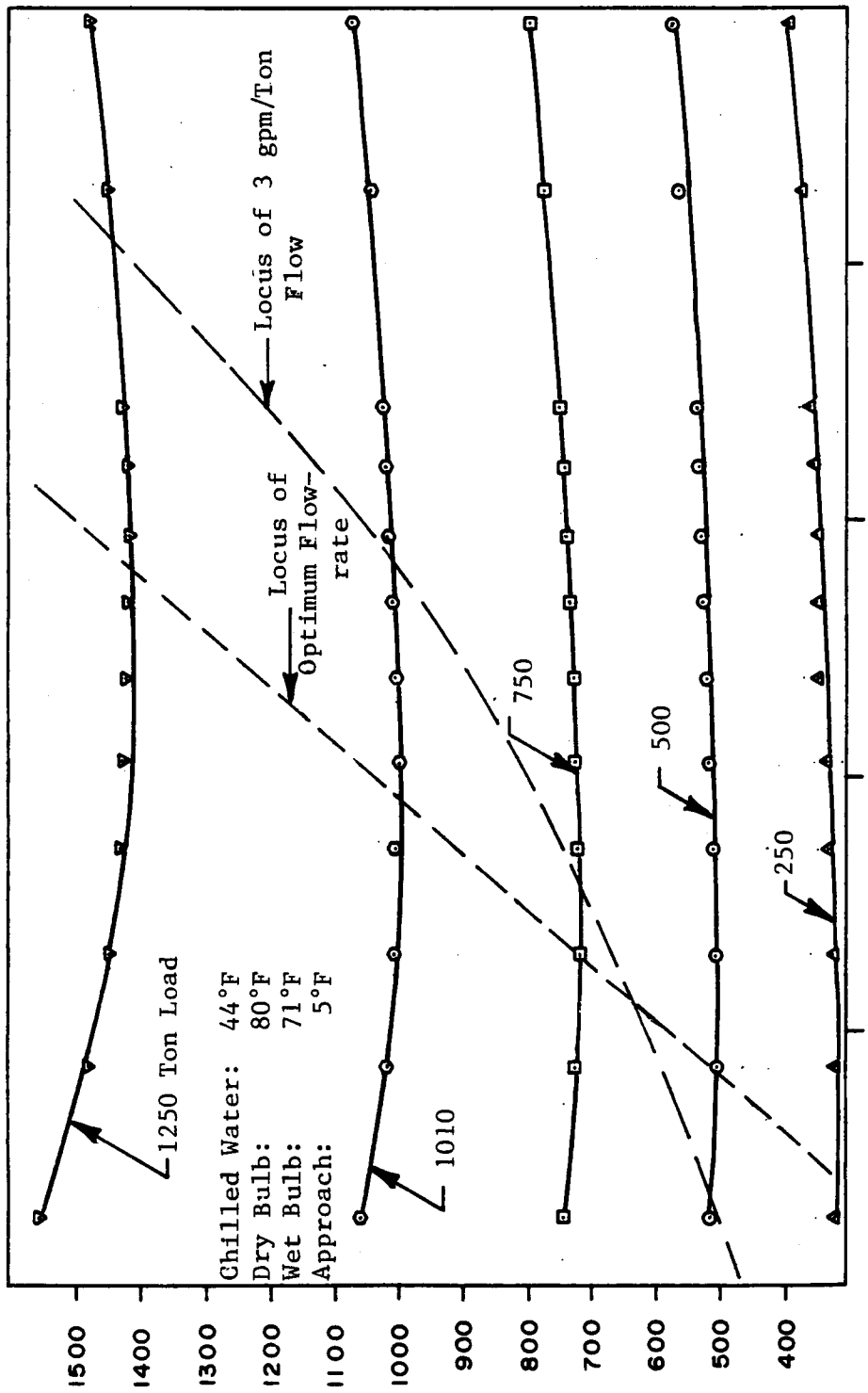


Figure 4.27. Effect of subsystem load on the optimum cooling water flowrate for Mode 3.3. (1250 ton electric in parallel with a 250 electric chiller.)

TABLE 4.3

ENERGY COST OF MODE 3.3 AS A FUNCTION OF THE COOLING WATER FLOWRATE*

	Load (tons)				
	1250	1000	750	500	250
Flowrate (GPM)			Cost (KWH)		
1455	1555	1063	748	514	320
1814	1477	1026	727	508	320
2088	1448	1013	724	509	324
2326	1434	1008	725	512	328
2537	1427	1007	729	516	332
2729	1424	1010	734	521	337
2906	1473	1015	739	526	342
3069	1424	1070	744	531	347
3222	1426	1026	750	537	353
3365	1429	1032	756	542	357
3867	1453	1056	778	563	377
4285	1478	1081	801	584	397

*Chilled Water Temperature: 44°F
Ambient Dry Bulb: 80°F
Ambient Wet Bulb: 71°F
Cooling Tower Approach: 5°F

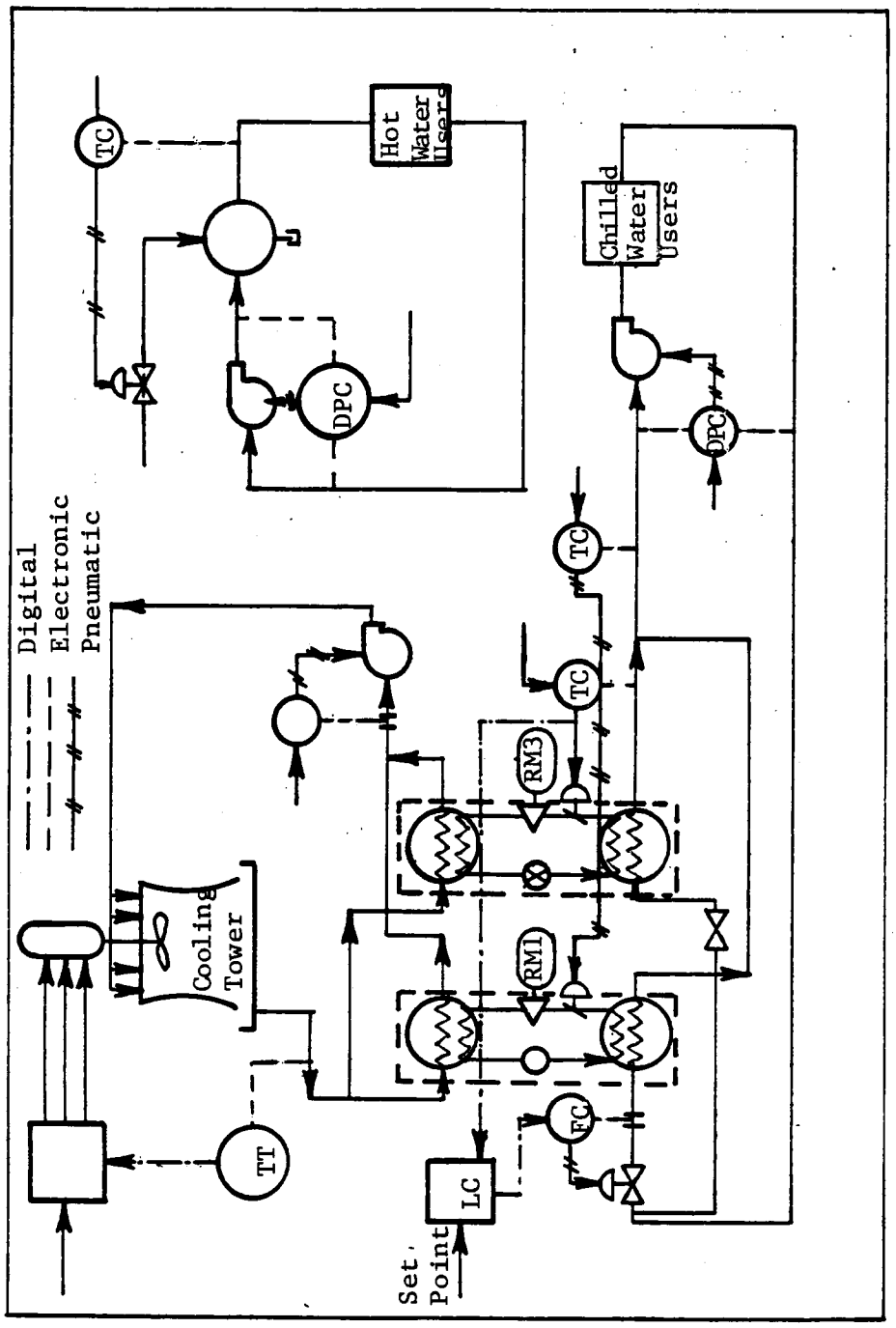


Figure 4.28. Mechanical refrigeration Mode 3.5.

cost, a set of studies have shown that for a constant electricity cost of 4.5 cents/KWH the cost of steam would have to be reduced by a factor of three (from $9/1000 lbs. to $3/1000 lbs.) before these optimization results are changed.

In Figure 4.29 a study of the operation of Mode 3.5 is presented for a 750 ton load with 44° F chilled water. The study explores the effect of shifting the load to the electric chiller upon the total equivalent energy cost of the submode. From this study one sees that the submode's optimum operation is when the small chiller is operated near its maximum capacity. The chilled water temperature supplied by RM-3 has little effect on the system energy cost. Figure 4.30 shows the effect of steam cost on the optimization of the submode. For $0.045/KWH electricity we see that a $9/1000 lbs. steam cost always favors the electric chiller's maximum operation. If steam is free, it is obvious the steam chiller is always favored. The data of Figure 4.30 indicates that steam would cost $3/1000 lbs. before the optimization problem is changed. At $3/1000 lbs. the system operation is most efficient with the small chiller operating at 150 KWH or 60% of its normal capacity.

We can conclude that the optimization of Mode 3.5 is quite straightforward. The cooling water flowrate can be a constant. In Figure 4.31 the cost curve is essentially unaffected by the cooling water flowrate. Table 4.4 suggests that a flowrate of approximately 3200 gpm will always be within 1% of the optimum for the load shown. Therefore the procedural optimization of the flowrate will set the cooling water flowrate at 3200 gpm. The smaller chiller should be operated to maximize its compressor work, and provide chilled water two degrees colder than the composite chilled water supply.

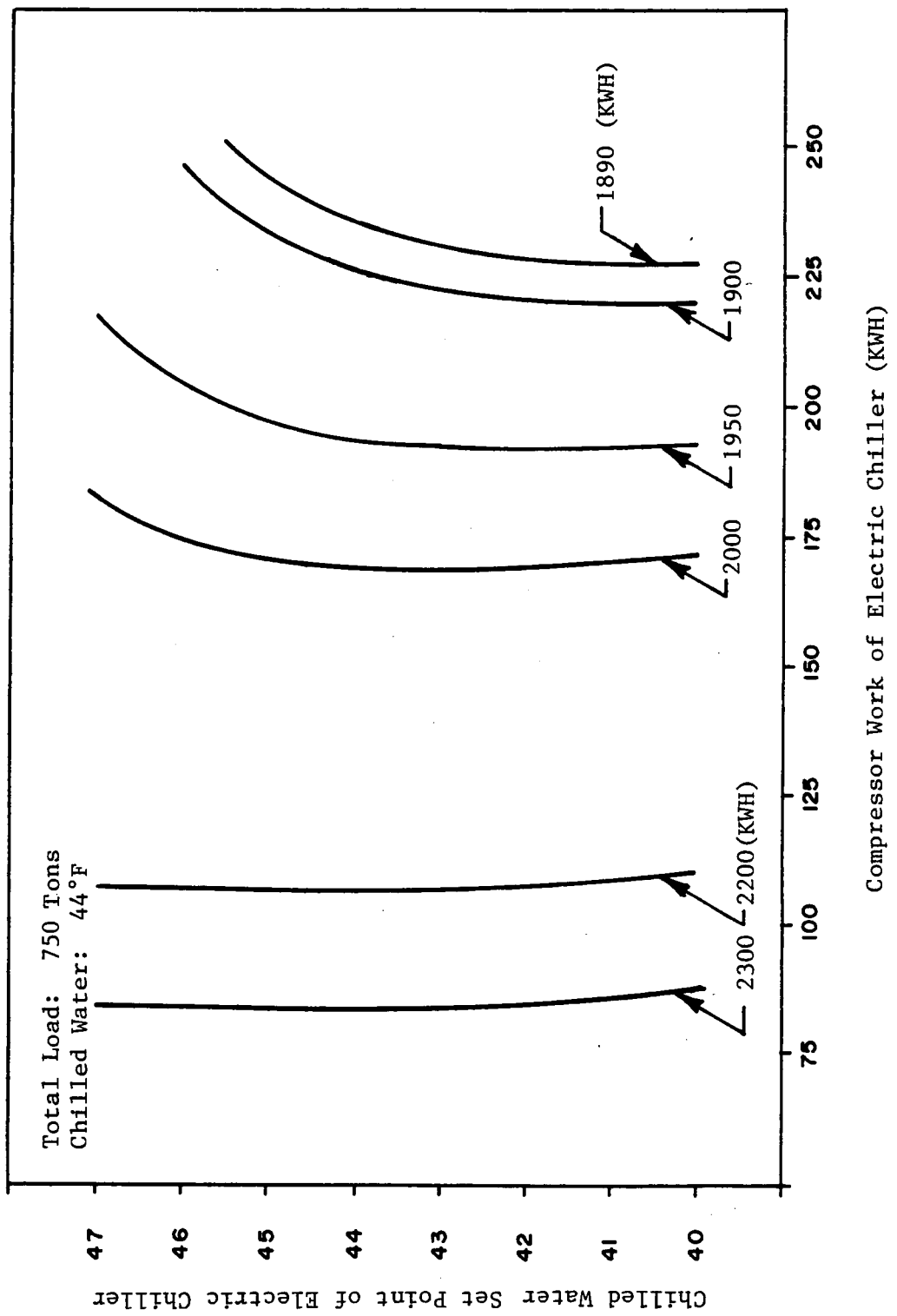


Figure 4.29. Constant cost contours as both chilled water supply and amount of load placed on small electric chiller are varied for Mode 3.5.

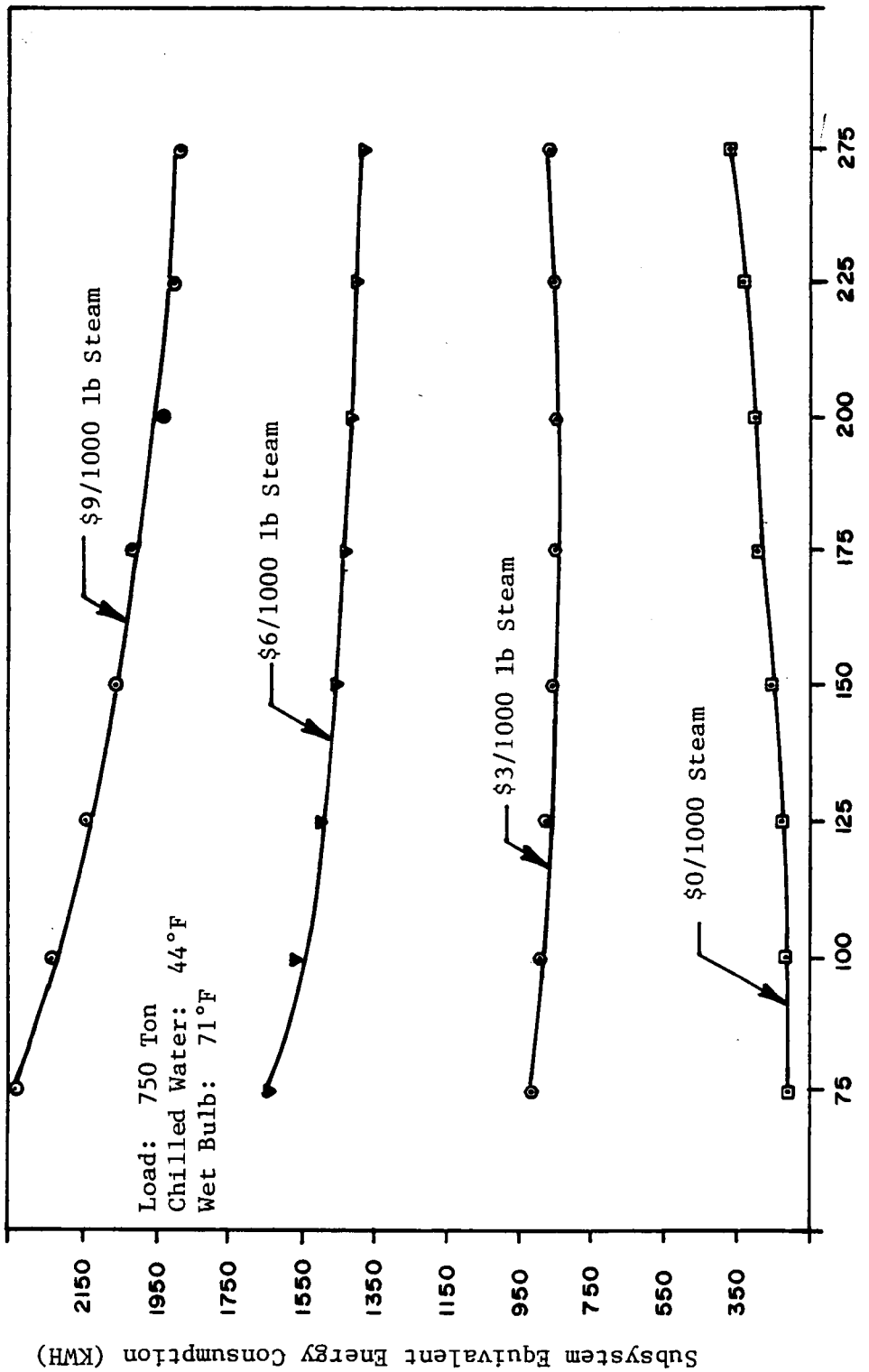


Figure 4.30. Effect of the unit cost of steam upon the optimum compressor loading of the small electric chiller for Mode 3.5. (Unit cost of electricity 4.5¢/KWH.)

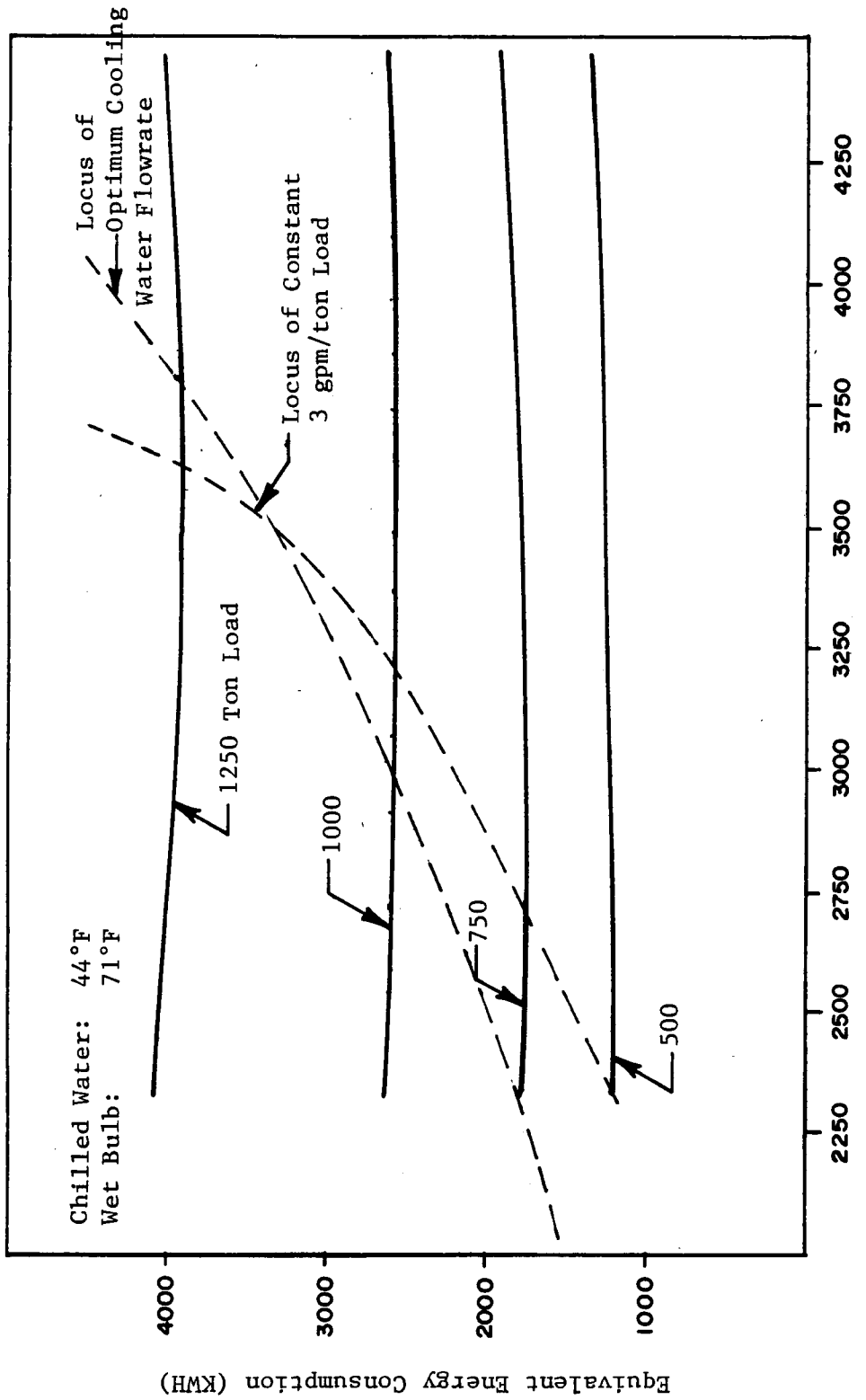


Figure 4.31. Effect of chilled water load on the optimum cooling water flowrate for Mode 3.5.

TABLE 4.4

ENERGY COST OF MODE 3.5 AS A FUNCTION OF THE COOLING WATER FLOWRATE*

	Load (tons)			
	1250	1000	750	500
Flowrate (GPM)	Cost (KWH)			
2322	4076	2736	1869	1224
2534	4042	2720	1863	1227
2726	4020	2710	1861	1231
2902	4006	2705	1862	1236
3065	3998	2702	1866	1240
3218	3992	2701	1871	1245
3361	3989	2702	1876	1250
3496	3980	2702	1881	1256
3624	3987	2704	1886	1261
3746	3988	2707	1892	1266
3862	3989	2710	1898	1271
3973	3990	2716	1903	1277

*Chilled Water Temperature: 44°F
Ambient Dry Bulb: 80°F
Ambient Wet Bulb: 71°F
Cooling Tower Approach: 5°F

Large Steam and Large Electric Chillers in Parallel Mode 3.6

In mode 3.6 both of the large chillers will be running. If this mode is ever applicable under normal operating conditions, its optimization is similar to that of Mode 3.5. The electric chiller's contribution to the total system cost should be maximized. The optimization of the cooling tower/cooling water loop is equivalent to the optimization in the other submodes. Figure 4.32 shows the system cost as cooling water flowrate varies. The data used in constructing this figure is in Table 4.4. From Table 4.5 the procedural optimization suggests a constant flowrate of 4000 gpm is appropriate. This flowrate will insure that the operational cost is always within 1% of the optimum cost.

After consideration of the characteristics of this mode it was concluded that Mode 3.6 would only be requested when:

1. Large system loads require it.
2. The smaller chiller is unoperable and one chiller cannot meet the system loads.
3. Under special peak electrical loading conditions.

If the operation of Mode 3.6 is mandated by conditions [1] or [2] the optimization routine should treat the mode the same as for Mode 3.5. The discussion of the special conditions of peak load will be deferred until Chapter VI.

Small Electric Chiller Mode 3.1

Mode 3.1 is similar to Mode 3.2 or Mode 3.4. In this submode the small chiller (RM-3) is operating alone to provide the chilled water needs of the building. This submode is not likely to be used often since the computer rooms most likely will require cooling which exceeds the

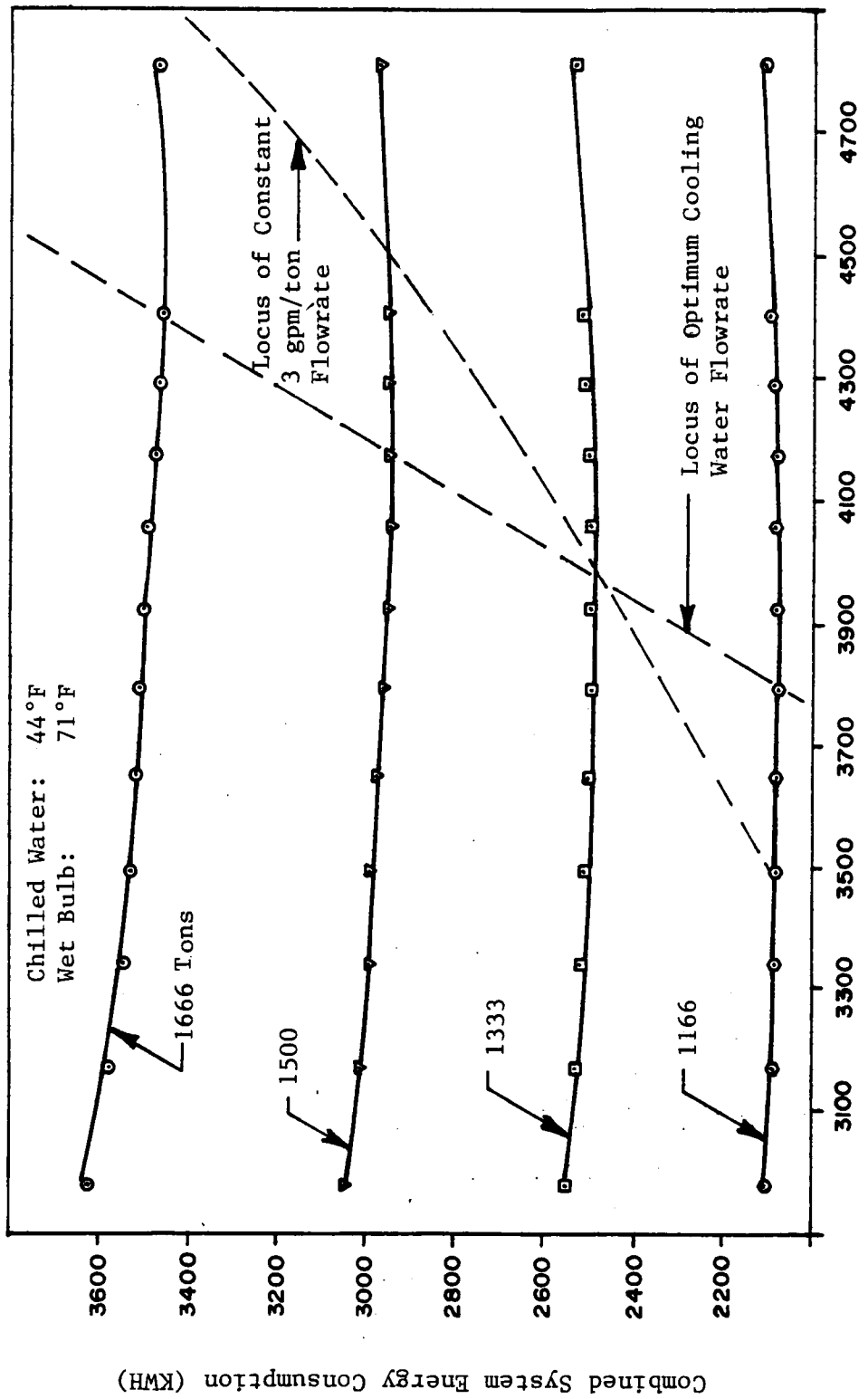


Figure 4.32. Effect of subsystem loading on the optimum cooling water flowrate for Mode 3.6. (1250 ton steam chiller in parallel with a 1250 ton electric chiller.)

TABLE 4.5

ENERGY COST OF MODE 3.6 AS A FUNCTION OF THE COOLING WATER FLOWRATE*

	Load (tons)			
	1666	11166	1333	1166
Flowrate (GPM)	Cost (KWH)			
2924	3616	3027	2551	2111
3162	3580	3004	2535	2098
3335	3553	2987	2524	2088
3498	3534	2974	2517	2081
3649	3519	2966	2512	2081
3792	3508	2959	2508	2078
3926	3500	2955	2506	2079
4055	3494	2952	2506	2085
4177	3489	2950	2506	2084
4294	3487	2950	2508	2088
4405	3485	2950	2513	2092
4810	3487	2957	2534	2111

*Chilled Water Temperature: 44°F
Ambient Dry Bulb: 80°F
Ambient Wet Bulb: 71°F
Cooling Tower Approach: 5°F

capacity of the machine. If the mode is ever requested the procedural optimization is exactly like that of Mode 3.2. The cooling tower approach temperature should be 5°F. The cooling water flow optimization is given in Figure 4.33 and in Table 4.6. A flow of 600 gpm will always be within 1% of the optimum.

Conclusion for Optimization of Mode 3

The results of the optimization studies of the Mechanical Refrigeration mode are quite straightforward. The studies indicate that a set of simple procedures for selecting operating conditions will assure that the energy cost will always lie within 3% of the optimum. One major optimization handle, the chilled water supply temperature, is assumed to be set by a feedback control system. For every degree increase in chilled water supply temperature, an energy savings of approximately 18 KWH of electricity is realized for an electric chiller. For a steam chiller with $9/1000 lbs. steam this savings is 46 KWH of equivalent electrical consumption. The proper operation of the cooling tower is also very important.

The results of the optimization study may be concisely summarized as:

1. Operate the cooling tower with a constant approach to ambient wet bulb.
2. Provide a constant (mode dependent) cooling water flowrate through the condenser of the chillers.
3. When two electric chillers are operating distribute the load so that the small chiller has 30% of the load or is operating at its maximum capacity if 30% exceeds its capacity.
4. When a steam and an electric chiller are operating under normal conditions, maximize the electric chiller's load.

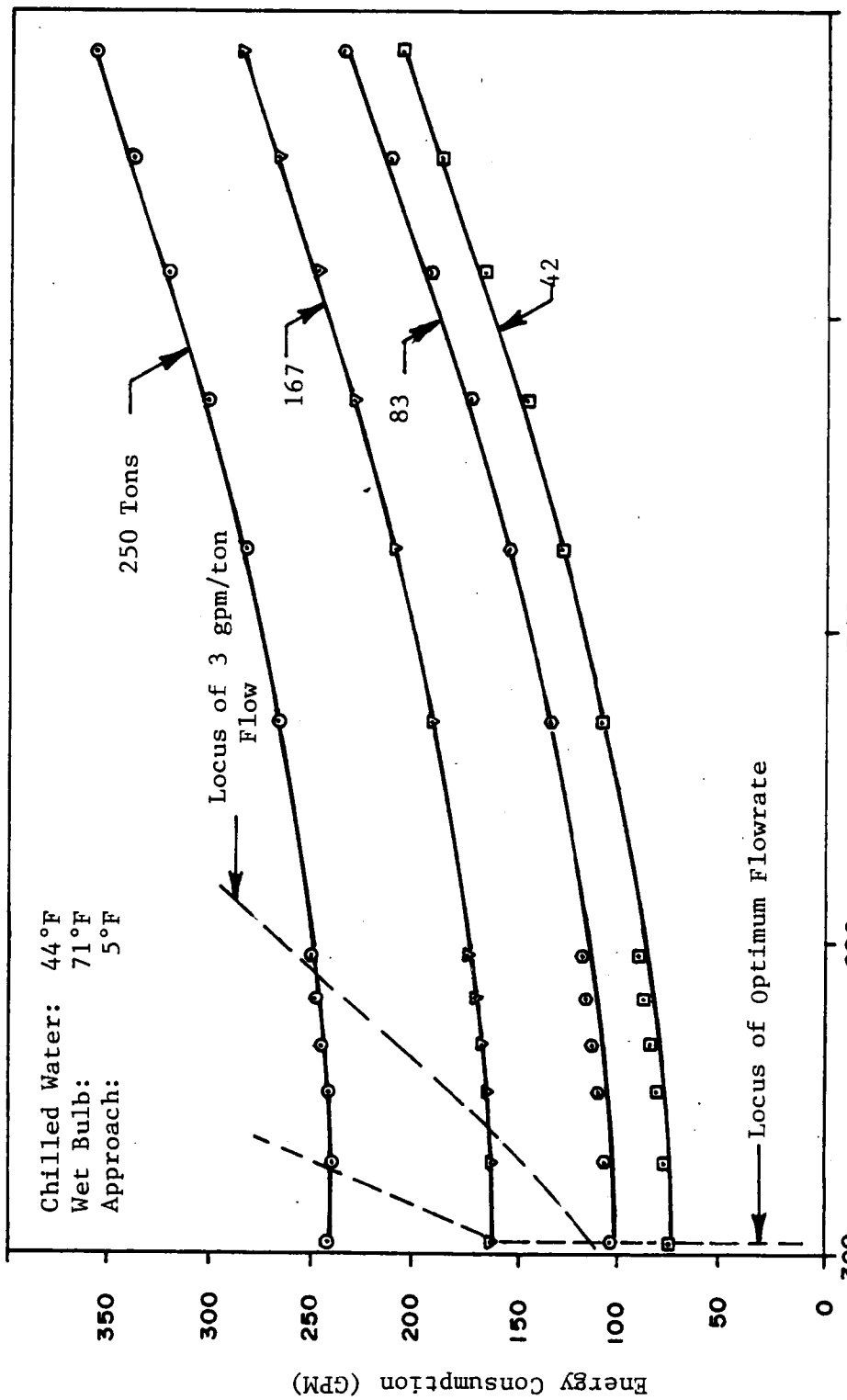


Figure 4.33. Effect of subsystem loading factor on the optimum cooling water flowrate of Mode 3.1. (250 ton electric chiller only.)

TABLE 4.6

ENERGY COST OF MODE 3.1 AS A FUNCTION OF THE COOLING WATER FLOWRATE*

	Load (tons)			
	250	167	83	42
Flowrate (GPM)	Cost (KWH)			
318	244	163	103	75
443	241	163	105	78
554	243	166	108	81
630	245	168	110	84
708	248	171	113	87
778	250	174	116	90
1150	267	192	135	109
1432	284	211	155	129
1668	302	230	174	149
1874	321	249	194	168
2059	339	268	214	188
2227	359	287	233	208

*Chilled Water Temperature: 44°F
Ambient Dry Bulb: 80°F
Ambient Wet Bulb: 71°F
Cooling Tower Approach: 5°F

5. When both electric chillers are running, the small chiller should provide chilled water 2°F colder than the total mixed chilled water temperature.

Other System Modes of Operation

There are twenty operational configurations which were studied. Since the previous discussions included both Modes 3 and 4, this leaves eight configurations which have not been discussed. This section will discuss general characteristics of each of these eight configurations.

Free Cooling Mode 1

In Figure 4.34 is a schematic of Mode 1. This mode uses one of the two electric chillers to provide low grade hot water to hot water users. The fraction of the total cooling load met by the heat pump is determined by the needs of hot water users. Any of the cooling load that is not provided by the heat pump is met by the cooling tower. In this mode the cooling tower pumps are in series with the chilled water pumps. The water strainer assures that the cooling tower water is of sufficiently high purity. Figure 4.34 shows that the flow ratio between the heat pump and the cooling tower is specified by the feedback loop of the hot water supply temperature controller. The chilled and hot water temperatures are assumed to be optimized by the feedback scheme of Figure 4.2, page 19. Since the chilled water supply temperature controller provides the set point to the cooling tower fan controller, the only optimization handle left for the supervisor is the heat pump's chilled water temperature. To minimize the interaction of the hot water and chilled water control loops, this should be set at the same temperature as the total mixed chilled water. The only optimization that may be performed is to determine whether the

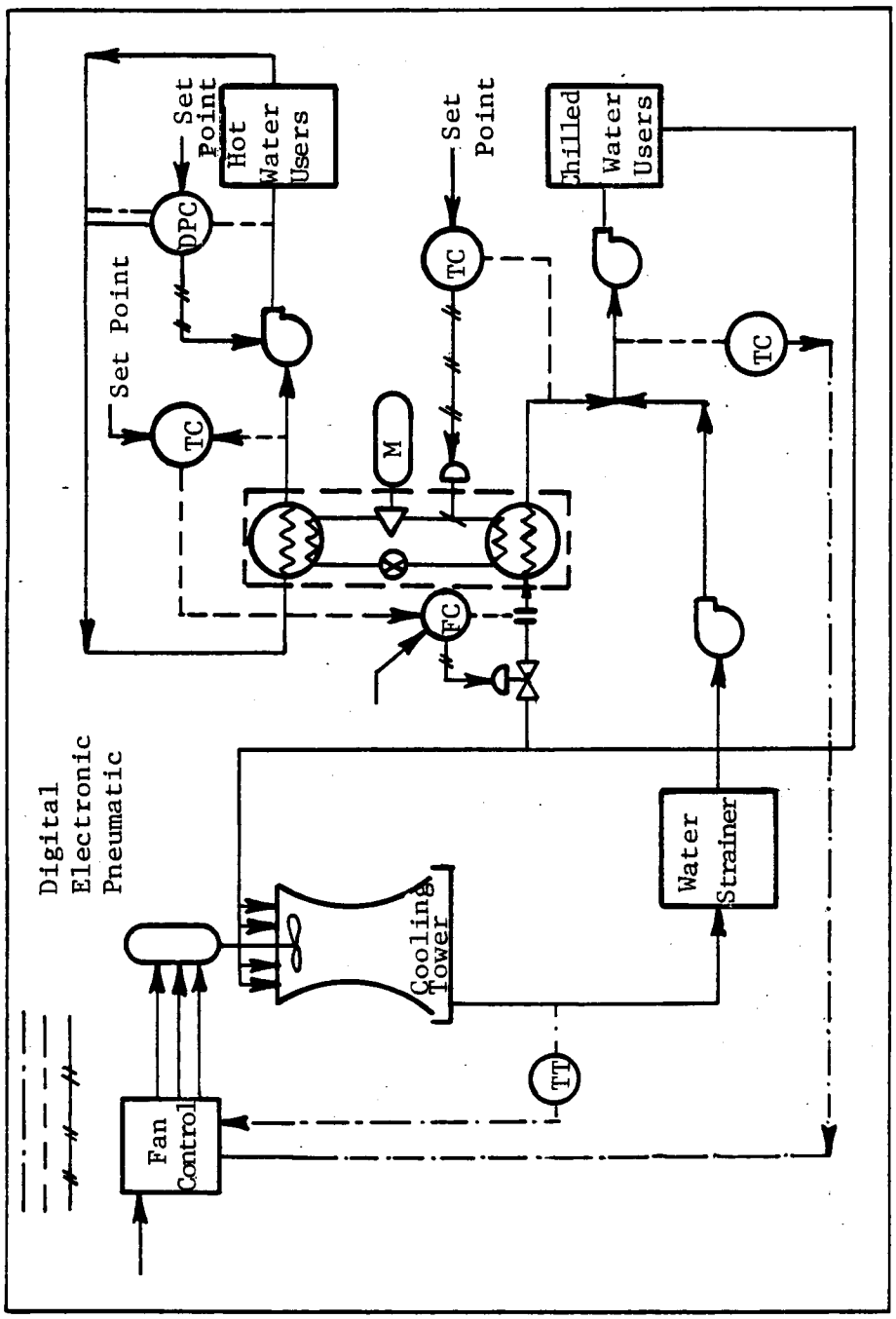


Figure 4.34. Free cooling with heat pump Mode 1. There are two submodes of operation. In Mode 1.8 the small chiller is configured as a heat pump and in 1.9 the large electric chiller is a heat pump.

mode is operational for the given set of operating conditions and if the mode is more cost effective than the other valid modes. This optimization is discussed in the next chapter.

Free Cooling with Steam Heat Mode 2

In Figure 4.35 is presented a schematic which is valid for either Mode 0 or Mode 2. The only difference in the two modes is whether the building has a heating load. Again, the primary water pumping networks establish conditions of the hot and chilled water loops and the optimization algorithm provides any other set points. The figure shows that no local equipment optimization handles are available for these modes. The optimization routine can only determine the validity and cost of the modes.

Mechanical Refrigeration with Heat Pump Mode 5

In Figure 4.36 is a schematic of Mode 5. This mode has two submodes. The small electric chiller is always configured as a heat pump while either the steam or large electric is configured as a chiller. The hot and chilled water conditions are again established by the feedback optimization. The cooling water optimization is the same for a single chiller in Modes 3.2 or 3.3. The hot water temperature control loop sets the flow distribution between the chiller and the heat pump. The heat pump's chilled water set point optimization is the same as for the operation of two parallel chillers. For all studies conducted this set point was two degrees below the composite mixed chilled water temperature. So the optimization task will be to determine the validity of the mode and its operating cost.

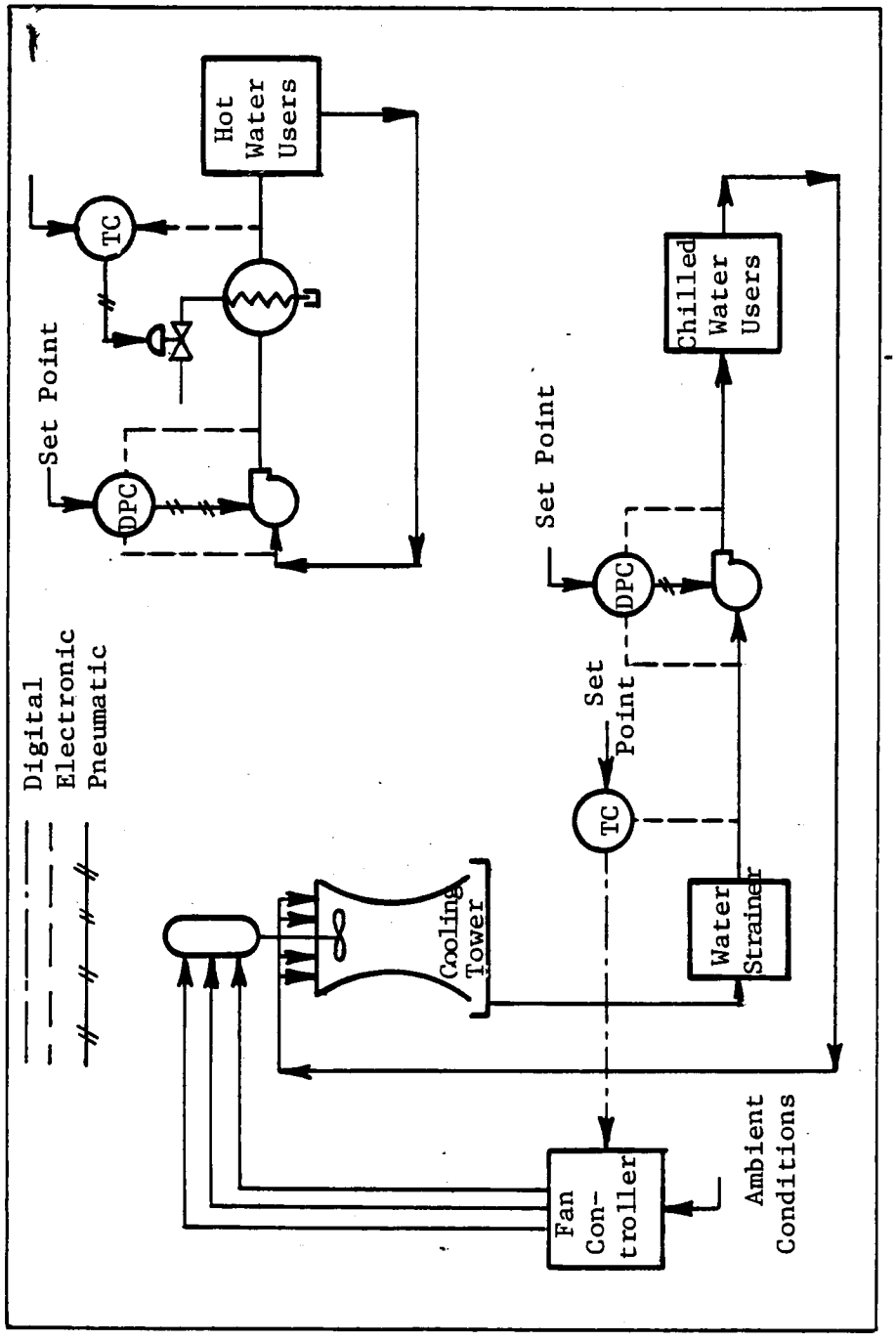


Figure 4.35. Free cooling (Mode 0) with steam heat (Mode 2).

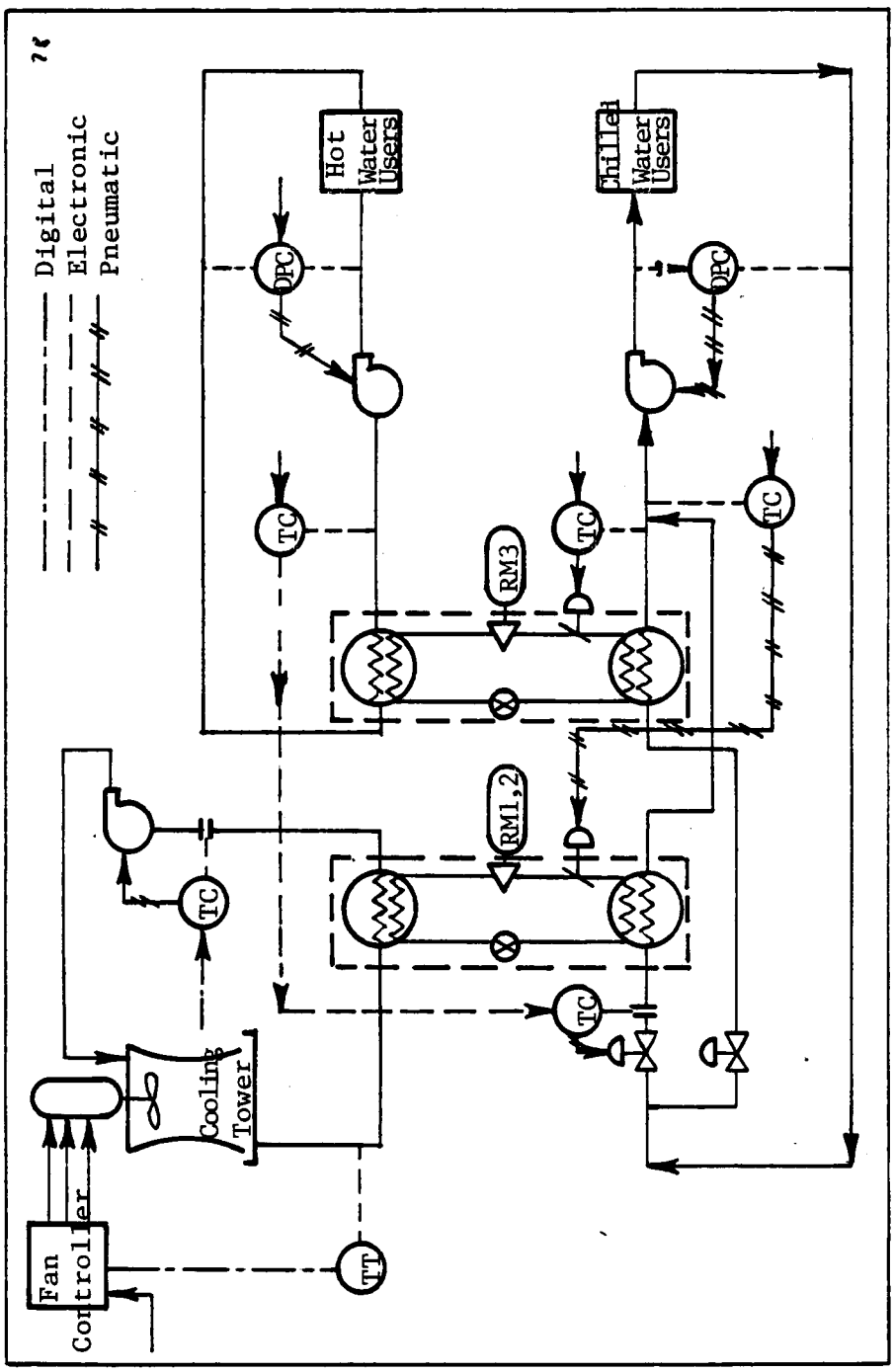


Figure 4.36. Mechanical refrigeration with heat pump Mode 5. This mode has two submodes of operation. The heat pump is always RM-3 (250 ton electric chiller) while the chiller is either RM-1 (1250 steam) or RM-2 (1250 electric).

Partial Heat Recovery Mode 6

The final configuration studied is the heat recovery Mode 6 (see Figure 4.37). There are two submodes for Mode 6. In this mode an electric chiller is configured to reject heat to the hot water users. Any excess energy is rejected by the cooling tower. The chilled water feedback optimization established the conditions at the evaporator. The hot water feedback optimization determines how much heat should be rejected to the cooling tower by manipulating the cooling tower bypass valve. The cooling tower supply temperature should be minimized to minimize the pressure drop across the pumping network. Therefore, this mode is also operated optimally and the supervisory optimization routine needs only to determine if the mode is valid.

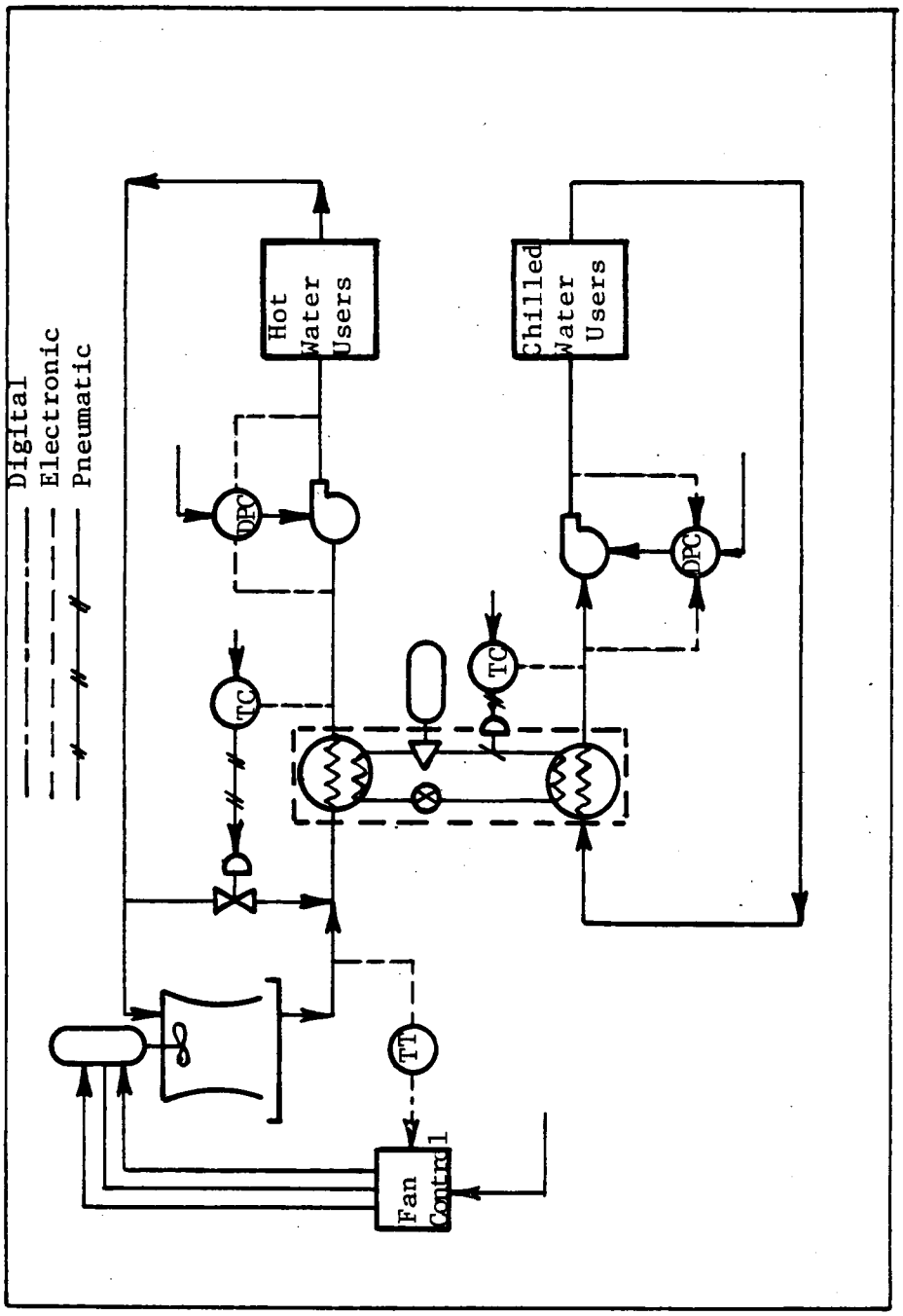


Figure 4.37. Heat recovery Mode 6. This mode has two submodes. In 6.1 the small chiller is on, in Mode 6.2 the large chiller is on.

CHAPTER V

ALGORITHMS FOR SELECTING THE OPTIMUM SYSTEM CONFIGURATION

In this chapter, a set of algorithms will be discussed which form the heart of the global optimization strategy for 590. The algorithms provide a mechanism for selecting the most cost effective primary water system configuration. The size and structure of these algorithms are appropriate for on-line implementation in the central operating station of the building. To facilitate the understanding of the algorithms and the methodology in employing them, most of the narrative will focus upon a schematic of each configuration and sections of a fortran subroutine developed to test the algorithms.

The algorithms are straightforward to implement and to use, and each system configuration is evaluated independently of the others. The main calling routine provides the algorithms with a set of variables which indicate the current status of the building's mechanical equipment (loads, supply temperatures, ambient conditions, and utility rates) and the evaluation routine determines which configurations are valid at these conditions and their cost for one hour of operation. The results from these evaluations can then be integrated with other pertinent system status information (equipment availability, projected duration of current conditions, peak load considerations, etc.) to select the appropriate operating configuration. The algorithms thus become the central focal point of the overall (global) optimization strategy for the building.

Scope of the Algorithms

The algorithms are basically a set of algebraic equations which make up a simplified empirical model. These simplified models have been developed to describe the major interactions revealed by the full scale simulation of the building. These equations use a set of input parameters to provide an estimation of equipment constraints and utility consumption. The estimations are based upon the optimal or near optimal operation of the configurations. By limiting the scope of the equations to predicting the cost of operation at the optimum, the set of equations (simplified models) contain no input terms raised to a higher power than two. The models were developed by performing detailed simulation studies over a wide range of operating levels, using the large scale simulator. The results of these studies were evaluated using hypothesized quadratic models whose coefficients were found using linear least squares curve fitting techniques. The criteria for accepting a particular form of a model was that it be as simple as possible while predicting the operating character of a configuration, or a particular component in a configuration, to within at least 10% of the simulation. The majority of the models are always within 5% of the simulation, but occasionally some of the models deviate from the desired goal. These deviations occur in configurations where the steam consumption of RM-1 has to be estimated at an operating point well below its anticipated operating range.

For the 13 configurations analyzed, the utility consumption of seven configurations is estimated directly by a single quadratic expression. This expression lumps the utility usage of all of the energy consuming devices together in one expression. Of the other six, some combination of a lumped parameter model and quadratic component models is required

to estimate the utility consumption. Component models were developed for:

1. All three chillers.
2. The electric chillers as heat pumps.
3. The cooling tower.
4. The pumping systems.

In several of the system configurations (Modes 1, 2, 5.11) the energy demand of each major component is predicted and the total usage computed as the sum of the individual demands. In other modes the consumption of a chiller is first estimated (steam required in the turbine) and the total electrical demand is predicted by a single expression (an example is Mode 3.4). The criteria for deciding which approach to take was based upon the difficulty in achieving the goal of 10% accuracy of the estimation.

Comparison of the Algorithms to the Full Scale Simulation

The models to be discussed in this chapter are easy to implement with an understanding of a few basic engineering principles. These algorithms are also very successful in predicting the operating character of the various system configurations. In Chapter VI there will be a discussion of the selection of the optimal mode of operation during the course of a typical day. In this chapter, it will be more appropriate to illustrate the accuracy of the models over a wide range of loading conditions.

In Figure 5.1 are plotted two hypothetical load profiles. The heating and cooling loads are both correlated to the ambient outside air temperature (16). The load on the chilled water system ranges from 3.5 million BTU's per hour at 50°F to 15.0 million BTU's per hour at 90°F.

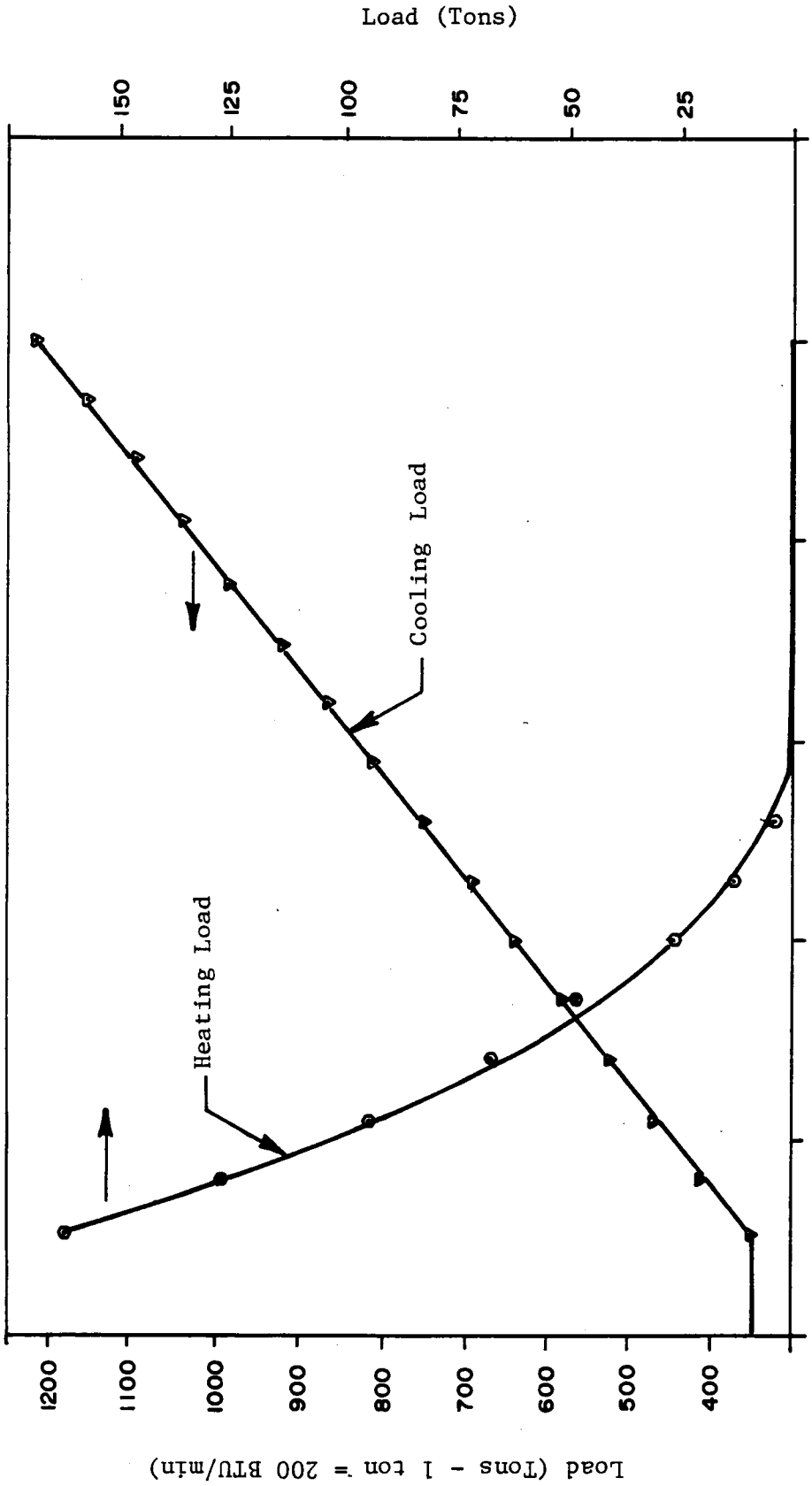


Figure 5.1 Hypothetical heating and cooling loads as outside air temperature varies.

The load on the hot water system ranges from 0 at 68°F to 7.5 million at 40°F.

In Figure 5.2 are plotted the operating costs and ranges of the system configurations as the outside air temperature is varied. This figure was generated using the full scale simulation. A similar set of curves, shown in Figure 5.3, were generated using the reduced model algorithms. From these two figures, it is readily seen that the reduced models performed well in comparison to the simulation. To provide a more detailed comparison, a sample datum point, one for 45°F, is shown in Tables 5.1 and 5.2. The only difference between the two tables is that the unit cost of electricity changes from $0.045/KWH in Table 5.1 to $0.100/KWH in Table 5.2. The unit cost of steam is unchanged in both tables.

From these tables one sees that all of the system configurations were correctly ordered, from least to most expensive, in Table 5.1 and only one mistake was made in Table 5.2. The percent loss for making this error would only be 8%, and this would only happen if the first four correctly ordered configurations were unavailable for implementation.

These are typical results; therefore, the simplified models are excellent predictors of the operating character of the water system. Furthermore, the subroutine which implemented these algorithms occupies 1/6 less program storage and executes 500 times faster than the program required to implement the full scale simulations.

It was suggested that IBM strongly consider implementing the algorithms which are to follow. The suggested procedure for implementation was to update the simulation to more closely reflect the actual

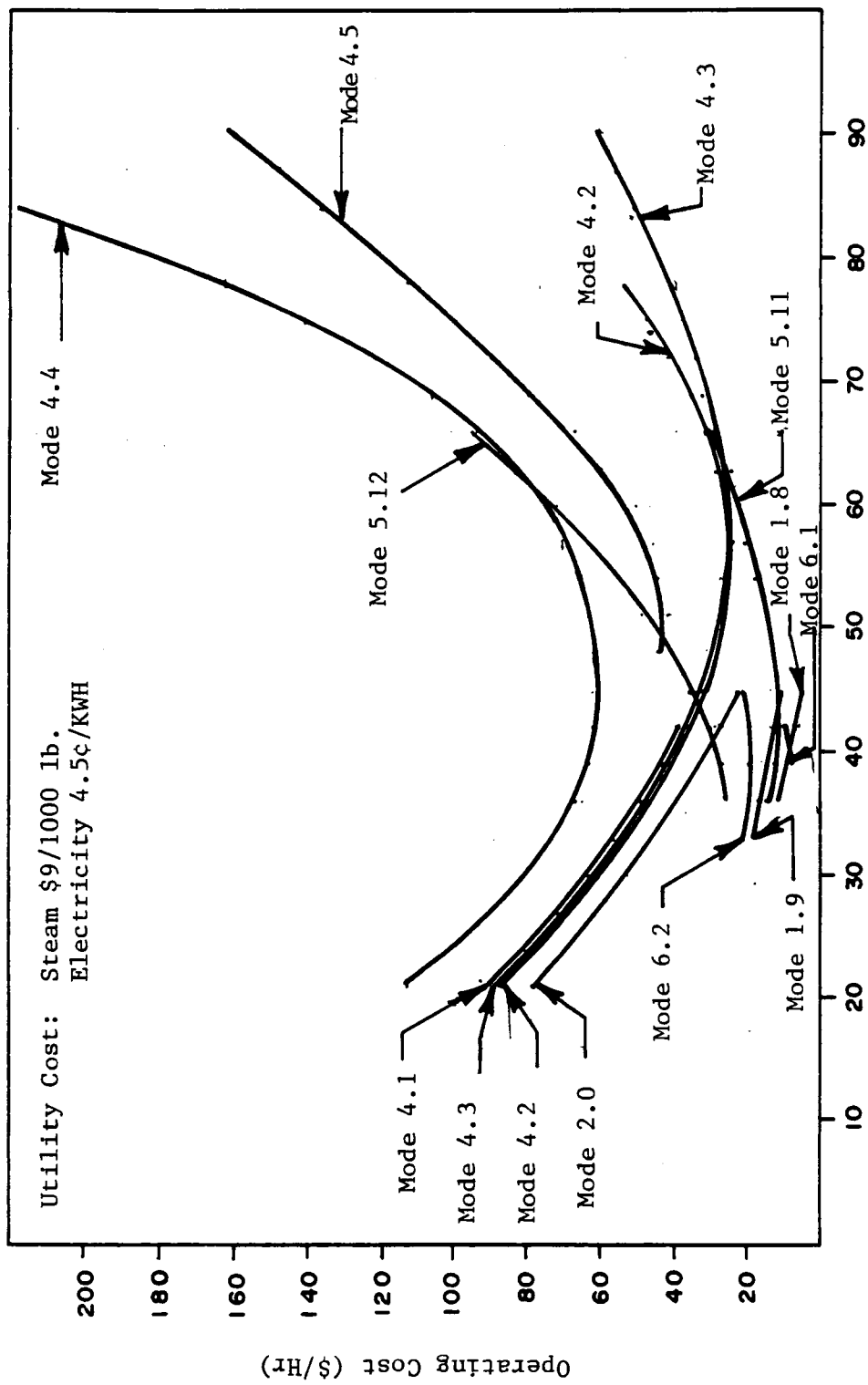


Figure 5.2. Operating cost of subsystem modes. Cost curves were generated from the comprehensive subsystem simulation. System heating and cooling loads are from Figure 5.1.

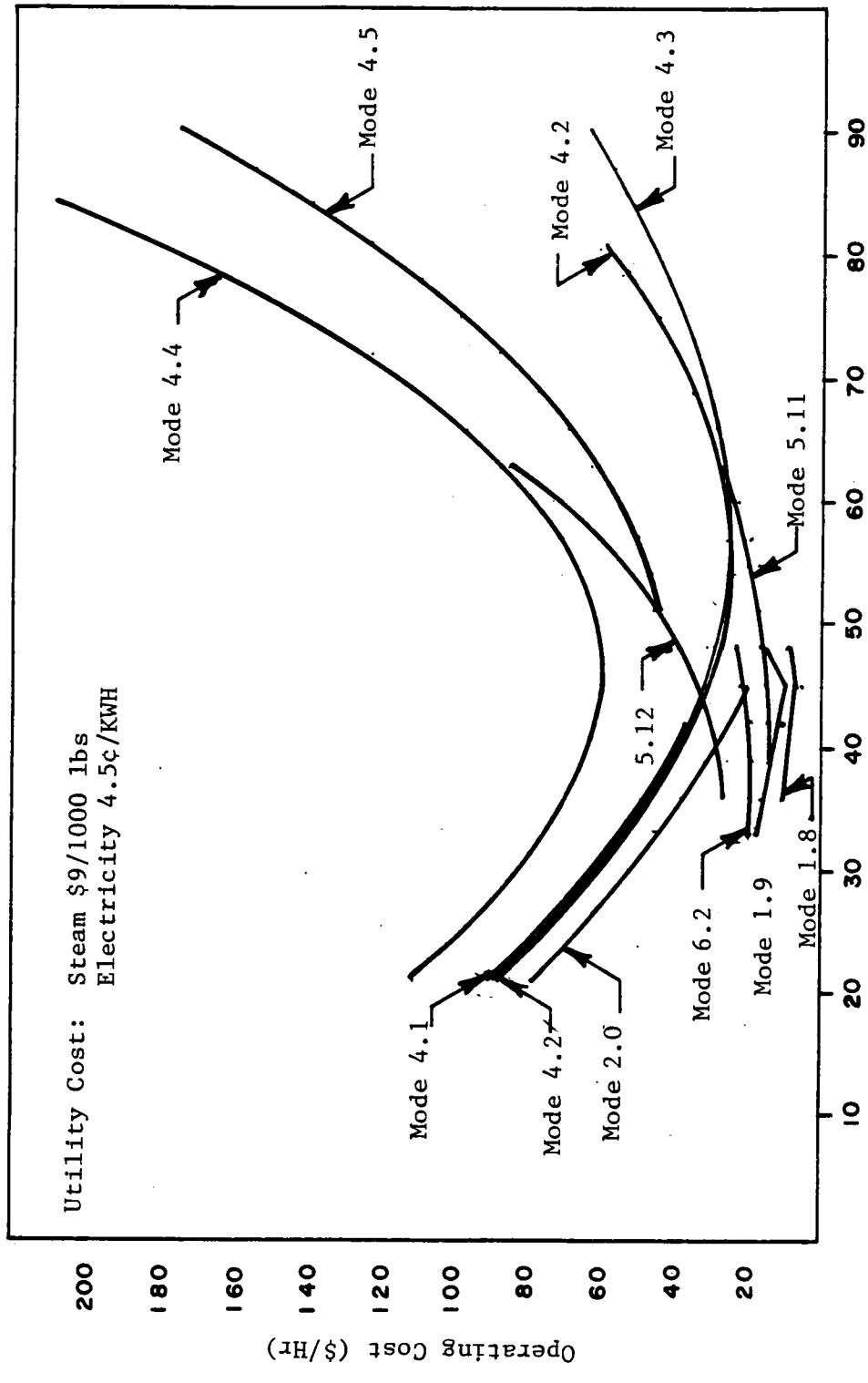


Figure 5.3. Estimated operating cost of subsystem modes. Curves were generated using the simplified mode models. System heating and cooling are from Figure 5.1.

TABLE 5.1

REPORT TO LIST THE APPLICABLE MODES OF THE WATER SYSTEM FOR AN ELECTRICAL RATE OF $0.045/KWH*

Mode	Submode	Estimated Cost %/Hr	Simulated Cost %/Hr	Percent Error of Estimates
1	8	6.74	7.50	-10.22
1	9	11.86	12.03	- 1.39
5	11	14.12	14.16	- 0.31
6	2	20.81	21.42	- 2.82
2	0	22.68	22.67	0.04
4	2	32.92	31.93	3.10
4	3	33.63	32.69	2.87
5	12	35.57	34.50	3.10
4	5	42.69	50.19	-14.94
4	4	62.05	58.79	5.55

*Outside Air Dry Bulb (DEG. F)	:	45.00
Outside Air Wet Bulb (DEG. F)	:	40.00
Chilled Water Load (MBTU/Hr)	:	4272.
Hot Water Load (MBTU/Hr)	:	1955.
Chilled Water Supply Temp. (DEG. F)	:	44.00
Hot Water Supply Temp. (DEG. F)	:	100.00
Chilled Water Temp. Rise (DEG. F)	:	8.00
Hot Water Temp. Rise (DEG. F)	:	10.00
Unit Cost of Electricity ($/KWH)	:	.045000
Unit Cost of Steam ($/MLB)	:	9.00

TABLE 5.2

REPORT TO LIST THE APPLICABLE MODES OF THE WATER SYSTEM FOR AN ELECTRICAL RATE OF $0.1/KWH*

Mode	Submode	Estimated Cost %/Hr	Simulated Cost %/Hr	Percent Error of Estimates
1	8	14.97	16.68	-10.22
2	0	25.10	25.08	0.07
1	9	26.35	26.73	- 1.39
5	11	31.37	31.47	- 0.31
6	2	46.25	47.60	- 2.82
5	12	47.46	43.80	8.36
4	2	47.85	45.65	4.82
4	3	49.45	47.36	4.41
4	5	59.13	67.01	-11.76
4	4	69.11	61.55	12.27

*Outside Air Dry Bulb (DEG. F)	:	45.00
Outside Air Wet Bulb (DEG. F)	:	40.00
Chilled Water Load (MBTU/Hr)	:	4272.
Hot Water Load (MBTU/Hr)	:	1955.
Chilled Water Supply Temp. (DEG. F)	:	44.00
Hot Water Supply Temp. (DEG. F)	:	100.00
Chilled Water Temp. Rise (DEG. F)	:	8.00
Hot Water Temp. Rise (DEG. F)	:	10.00
Unit Cost of Electricity ($/KWH)	:	.100000
Unit Cost of Steam ($/MLB)	:	9.00

operating character of the building, then to use the simulation to update the coefficients in the quadratic models of the algorithms.

Discussion of the Algorithm

The discussion of the algorithms will focus primarily upon a Fortran IV subroutine (MODS) developed to test and implement the reduced models. It is expected that the global optimization supervisor will provide to this subroutine a set of major system variables which define the current status of the primary water system. Using this input set, the routine compiles a list of configurations which are valid at the current conditions. The routine also returns the estimated utility consumption for each configuration for an hour of operation. It is the supervisor's responsibility to factor in all other pertinent considerations required in the selecting of the optimal mode of operation.

A list of arguments which are used by the reduced models and their functional definitions is given in Table 5.3. These arguments correspond to the variables in the calling argument of MODS, seen in Figure 5.4.

Several of the arguments in Table 5.3 require further explanation. For instance, the variable TSET (variable 4) is the temperature of the small chiller's (RM-3) chilled water set point in Modes 3.3 and 3.5, while it is the heat pump's (RM-2 or RM-3) in Modes 1 or 5. The variable CRATIO is the cost ratio of steam to electricity. This is used to convert the estimated steam usage in the various configurations into equivalent kilowatt hours.

Finally, the arrays IVMOD and CMOD are used to communicate which models are valid and the energy consumption of the valid modes. When control is returned to the calling routine, IVMOD contains an array of 0's

TABLE 5.3

INDEPENDENT VARIABLES PROVIDED TO MODS

Variable	Units	Description
1. QCHUD	BTU/Hr	Total load on the chilled water system
2. QHWUD	BTU/Hr	Total load on the hot water system
3. TCHSD	DEG. F	Chilled water supply temperature
4. TSET	DEG. F	Chilled water temperature supplied by heat pumps in Modes 1 and 5
5. THWSD	DEG. F	Hot water supply temperature
6. DTCH	DEG. F	Temperature rise across the chilled water system
7. DTHOT	DEG. F	Temperature drop across the hot water system
8. TWBOA	DEG. F	Ambient wet bulb temperature
9. CRATIO	KWH/lb	Ratio of steam cost to cost of electricity
10. IVMOD	-	Array which holds the information concerning which modes are valid
11. CMOD	KWH	Array which holds the estimated energy usage for each configuration

```
      SUBROUTINE MODS(QCHUD,QHWUD,TCHSD,TSET,THWSD,DTCH,DTHOT
     1 ,TWBOA,CRATIO,IVMOD,CMOD,IJMP)
      DIMENSION IVMOD(12),CMOD(12),TDAT(21),FCOF(21),FCOF1(21)
      DIMENSION DPMAX(3),FCOEF(3),FMAX(3)
      DATA W3MAX/270.0/,W2MAX /1250.0/,W1MAX/1404/
      DATA W3MIN/20.0/,W2MIN/80.0/,W1MIN/50.4/
      DATA THMAX/105.0/
      DATA DPMAX/82.370,86.70,86.70/
      DATA FCOEF/2.04,2.04,2.04/
      DATA FMAX/.4002E7,2.816E6,1.686E6/
      DATA HSTM/850.0/
      DATA FCOF1 /4.961,0.0,0.0,0.0,-5.622,3.904,0.0,-0.409
     1 ,0.0,-15.677,0.0,0.0,0.0,0.0,0.0,0.0,0.0,0.0,0.0,-2.933
     2 ,14.805,.73175/
C
C
C
C     SCALE INPUT VARIABLE AROUND THEIR NOMINAL
C     DESIGN VALUE;  These variables are defined in Table 5.6
C
C
      QC=QCHUD/15.0E6 !SCALE COOLING LOAD
      QH=QHWUD/10.0E6 !SCALED HEATING LOAD
      TCHS=TCHSD/42.0 !SCALED CHILLED WATER SUPPLY TEMP
      TSETX=TSET/42.0 !SCALED HEAT PUMP SUPPLY TEMP
      THWR=(THWSD-DTHOT)/90.0 !SCALED CONDENSOR INLET TO CHILLERS
      DTC=DTCH/12.0 !SCALED DELTA TEMP OF CHILLED WATER
      DTC1=((TCHSD-TSET)+DTCH)/12.0 !DELTA T OF HEAT PUMP
      DTH=DTHOT/12.0 !DELTA TEMP OF HOT WATER USERS
      DWB=TWBOA/80.0 !SCALED AMBIENT WETBULB TEMP.
      TDWB=DWB
      IF(TWBOA.LT.50.0)TDWB=55.0/80.0
```

Figure 5.4. Initialization section of subroutine modes.

or 1's. A zero indicates that the configuration associated with that location is not operational for the given input conditions. The correspondence between locations in IVMOD and CMOD is given in Table 5.4.

Internal Variables

Several variables are used internally by the algorithms in MODS. These variables are designated by Fortran data statements in Figure 5.4. A description of these variables and their usage is provided in Table 5.5.

Scaled Variables

Once the subroutine MODS begins execution, the first few steps are used to scale the input variables that the calling routine has passed to MODS. The reason for scaling these variables is so the coefficients in the quadratic models are all of the same order of magnitude.

The scaled cooling load QC is computed by dividing the cooling load QCHUD by 15.0 million btu's/hr. This scaling factor was chosen because it is the nominal capacity of the chilled water system when RM-2 and RM-3 are running in parallel, with 80°F cooling water available. Sometimes a particular model rescales QC. This occurs when the expression is predicting the compressor work of the small chiller (RM-3). The nominal capacity of this chiller is 3.0 million btu's.

There is also a scaled heating load QH. This variable is created by dividing the heating load QHWUD by 10.0 million. QH is also rescaled occasionally when a particular expression deals exclusively with RM-3.

In Table 5.6 is a listing of all the scaled variables used by the quadratic models. Figure 5.4 shows the initialization section of the subroutine.

TABLE 5.4

RELATION OF A CONFIGURATION TO ITS LOCATION IN IVMOD OR CMOD

Location	Configuration Number
1	Mode 1.8
2	1.9
3	2.0
4	3 or 4.1
5	3 or 4.2
6	3 or 4.3
7	3 or 4.4
8	3 or 4.5
9	5.11
10	5.12
11	6.1
12	6.2

TABLE 5.5

INTERNAL VARIABLES

Variable	Description
W1MAX	Maximum compressor work of RM-1
W2MAX	Maximum compressor work of RM-2
W3MAX	Maximum compressor work of RM-3
W1MIN	Minimum compressor work before the hot gas bypass is activated
W2MIN	
W3MIN	
THMAX	Maximum safe condensor exit temperature for chiller
DPMAX	Maximum pressure drops for the pumping network at full flow
FCOEF	Used in estimating the pump efficiencies
FMAX	Maximum pumping rates in the pumping systems (lb./hr)
HSTM	Latent heat of condensation of steam

TABLE 5.6

DEFINITION OF THE SCALED VARIABLES USED BY THE QUADRATIC MODELS

Scaled Variable	Corresponding variable(s) in Subroutine Argument	Definition
QC	QCHUD	Scaled chilled water temperature
QH	QHWUD	Scaled hot water heating load
TCHS	TCHSD	Chilled water supply temperature
TSETX	TSET	Scaled heat pump supply temperature
THWR	THWSD,DTHOT	Condensor inlet temperature
DTC	DTCH	Temperature rise across the chilled water
DTC1	TCHSD,TSET,DTCH	Temperature drop across a heat pump
DTH	DTHOT	Temperature drop across the hot water system
DWB	TWBOA	Ambient wet bulb
TDWB	TWBOA	Scaled wet bulb used in Mode 3

Discussion of the Individual Configurations

Once MODS has finished scaling the input parameters, it begins evaluating the individual system configurations. The first mode is Mode 1. In this mode the cooling tower and a heat pump are running in parallel to supply the chilled water needs of the building. The heat pump is configured as a total heat recovery device meeting all of the low-grade hot water needs of the building.

Free cooling with heat pump Mode 1. Mode 1 has two submodes. The first submode, Mode 1.8, has the small electric chiller (RM-3) configured as a heat pump. In Mode 1.9 the large electric chiller (RM-2) is configured as the system heat pump. In Figure 5.5 is a flow chart which provides an overview of the computations performed in evaluating the submodes.

The algorithms begin by first estimating the efficiency of the heat pump. This efficiency is in terms of the Coefficient of Performance (COP). The COP is defined as the ratio of the evaporator load and compressor work. The quadratic model used to estimate the COP of RM-3 in Mode 1.8 is shown in Figure 5.6. (This figure is from a section of the subroutine MODS.) The COP was found to be readily described as a function of the hot water load (QH), the heat pump's chilled water set point (TSEXT), the hot water temperature set point (THWR), and the temperature change (DTH and DTC1) across the condensor and the evaporator. The compressor work and the corresponding cooling load needed to match the heating load (QH) are easily calculated using the input heating load and the estimated COP. The compressor work is given by the expression:

$$\text{Compressor work} \doteq \text{cooling load}/\text{COP} \qquad (1)$$

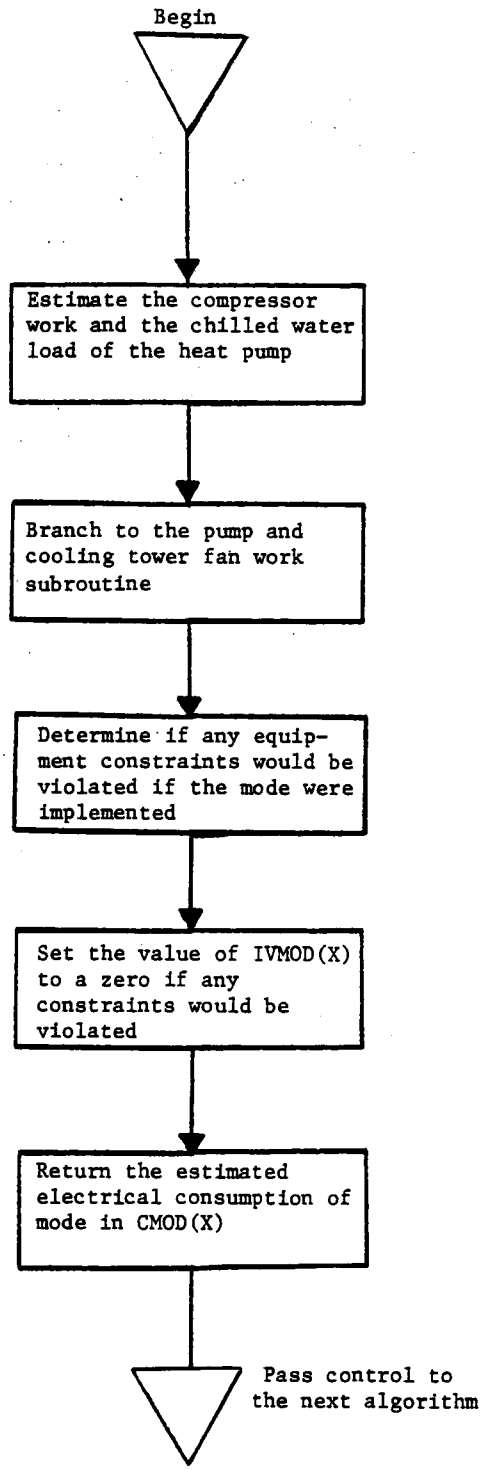


Figure 5.5. Flow chart of computational logic used in the Algorithms of Mode 1.

```
C--MODE 1-8 FIRST
C--FREE COOLING WITH RM3 AS A TOTAL HEAT RECOVERY DEVICE
C
C--ESTIMATE THE COMPRESSOR COP
       COP=2.307*QH*(10.0/3.0)+0.6002*TSETX-1.108*THWR
       1 -0.0670*DTH-0.0177*DTC1-1.269*((QH*(10.0/3.0))**2)
       2 +0.85959
C
       COP=3.0*COP
C
C-----ESTIMATE THE HEAT PUMP COMPRESSOR WORK
C
       Q3CH=(COP*QHWUD)/(1.0+COP)
       QCH=Q3CH
       W3COMP=(QHWUD-Q3CH)/3412.0
C
C-----FINALLY COMPUTE FREE COOLING CONTRIBUTION TO TOTAL COST
       IRET=1
       GOTO 1000
10     CONTINUE
C
C-----DETERMINE THE VALIDITY OF THE MODE
       IVMOD(1)=1
       IF(W3COMP.GT.W3MAX)IVMOD(1)=0
       IF(W3COMP.LE.0.0)IVMOD(1)=0
       IF(QCT.LE.0.0)IVMOD(1)=0
       IF(TCTR.LT.(TWBOA+2.0))IVMOD(1)=0
       IF(THWSD.GT.THMAX)IVMOD(1)=0
C-----RETURN TH ESTIMATE OF THE COST IN KWH
       IF(W3COMP.LT.W3MIN)W3COMP=W3MIN
       COST=WCTS+W3COMP+PCOST
       CMOD(1)=COST
```

Figure 5.6. Algorithm for Mode 1.8.

The heat rejected by the heat pump is given by:

$$\text{Heat rejected} = \text{cooling load} + \text{compressor work} \qquad (2)$$

Substitution of equation (1) into (2) and rearranging yields an equation which related the cooling load of the heat pump to the heating load and the COP. This expression is:

$$\text{Cooling load} = (\text{heating load} \times \text{COP})/(1 + \text{COP}) \qquad (3)$$

Equation (3) is the expression used in Figure 5.6 to estimate the small chiller's cooling load, Q3CH. The compressor work, W3COMP, is then calculated and converted to kilowatts.

With the compressor work and the heat pump's evaporator cooling load estimated, the next task is to determine the work required in the pumping networks and the fan work of the cooling tower. This is accomplished by a subroutine that is common to Modes 1.8, 1.9, 2.0, and 5.11. A flowchart of the computational logic of this subroutine is given in Figure 5.7.

In Figure 5.8 is the fortran code used in evaluating the fan and pumping work. The subroutine entry point is at line 1000 for the Mode 1 algorithms. The pumping algorithm begins by calculating the total chilled water flowrate from the given load and temperature rise (DTCH). The portion of the total flow that goes through the heat pump is then calculated from the estimated heat pump cooling load and the given temperature drop (DTCH1) across the heat pump's evaporator. The flowrate through the cooling tower is the total flow minus the flow through the heat pump.

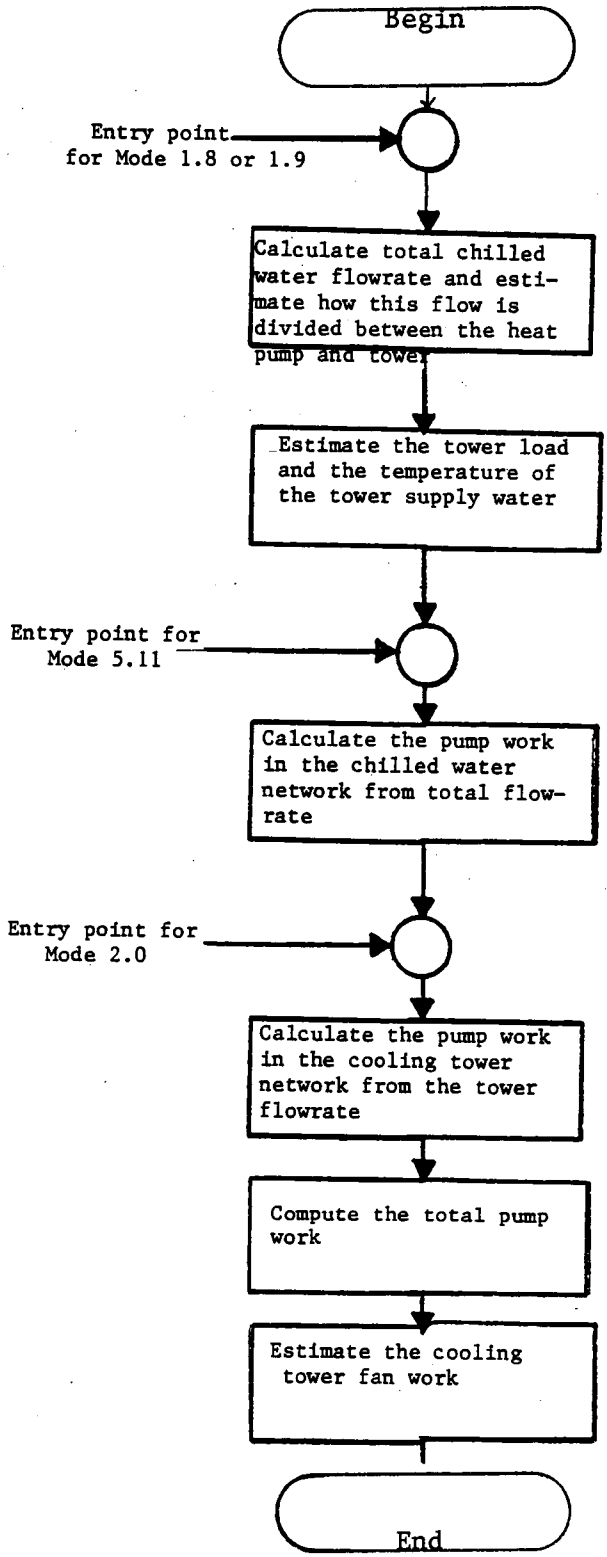


Figure 5.7. Flow chart of computational logic of the pump and cooling tower fan work subroutine.

```
C-----ENTRY FOR MODE 1-X
C
1000    CONTINUE
C
        WFLOT=QCHUD/DTCH          !CHILLED WATER FLOW RATE
        WFCH=QCH/(DTC1*12.0)      !CHILLER CHILLED WATER FLOW
        WFCT=WFLOT-WFCH           !COOLING TOWER FLOW RATE
C
        QCT=QCHUD-QCH             !LOAD ON THE COOLING TOWER
        TCTR=TCHSD+DTCH-(QCT/WFCT) !COOLING TOWER SET POINT
C-----ENTRY FOR MODE 5-11
1020    TDXT=42.0
C
C------CALCULATE PUMP WORK FOR CHILLED WATER PUMPS
C------ALL WATER IS PUMPED THROUGH THE CHILLED WATER PUMPS
C---estimate the pressure drop in the chilled water network
        DPCHP=0.0+DPMAX(2)*(WFLOT/FMAX(2))**FCOEF(2)
        FR=WFLOT/FMAX(2)
C----estimate the efficiency of the pumps
        EFF=FR*(1.80088+FR*(-0.83846-0.151531*FR))+1.0E-4
C----calculate the pump work in the chilled water network
        WCHPMP=0.7457*(WFLOT*DPCHP)/(1714.0*8.34*60.0*EFF)
C------ENTRY FOR MODE 2-0
C
1001    CONTINUE
C-------CALCULATE THE COST OF COOLING TOWER PUMPS
C------ pressure drop in CT network
        DPTWR=10.0+DPMAX(1)*(WFCT/FMAX(1))**FCOEF(1)
        FR=WFCT/FMAX(1)
        EFF=FR*(1.8008+FR*(-0.83846-0.151531*FR))+1.0E-4
        WCTPMP=0.7457*WFCT*DPTWR/(1714.0*8.34*60.0*EFF)
C
C------COMPUTE THE ESTIMATE OF THE TOTAL PUMP COST
        PCOST=WCTPMP+WCHPMP
```

Figure 5.8. Algorithm for pumping and cooling tower fan work.

```
C-----NOW ESTIMATE THE COOLING TOWER FAN COST
      QC1=QCT/15.0E6
      CTA=(TCTR-TWBOA)/15.0    !COOLING TOWER APPROACH
      TCHST=TCTR/TDXT
C
      TDAT(1)=QC1
      TDAT(2)=TCHST
      TDAT(3)=DTC
      TDAT(4)=DWB
      TDAT(5)=CTA
C----COMPUTE FREE COOLING COOLING TOWER FAN COST
      ICOUNT=6
C
      DO 1005 I=1,5
      DO 1005 J=I,5
      TDAT(ICOUNT)=TDAT(I)*TDAT(J)
      ICOUNT=ICOUNT+1
1005  CONTINUE
C
      WCTS=0.0
      DO 1010 I=1,20
      WTST=FCOF1(I)*TDAT(I)
      WCTS=WCTS+WTST
1010  CONTINUE
      WCTS=WCTS+FCOF1(21)
      WCTS=57.88*WCTS
      IF(WCTS.LT.0.0)WCTS=3.0
      IF(CTA.GT.0.5)WCTS=3.00
C
      GOTO(10,20,30,95)IRET
```

Figure 5.8. (Continued)

The cooling tower load is computed as the total load minus the load required at the heat pump. (Note that this load will be negative if the heating load exceeds the capacity of the chilled water system.) If the tower is operating at a different supply temperature than the heat pump, the composite chilled water temperature is obtained by mixing the two streams. Thus the temperature of the chilled water supplied by the tower must be estimated. The required cooling range of the tower is known from its load and the flowrate of water through it. This range is calculated as:

$$\text{Cooling range} = \text{tower load/flow through the tower} \qquad (4)$$

The temperature of the water returning to the chilled water system is calculated from input parameters in the calling argument of MODS. This temperature is given by the equation:

$$\text{Return temperature} = \text{TCHSD} + \text{DTCH} \qquad (5)$$

The tower supply temperature, TCTR in Figure 5.8, is the return chilled water temperature minus the tower cooling range.

After determining the system flowrates, the tower load, and the tower supply temperature, the next task is to estimate the pumping work required in the chilled and cooling tower pumping networks. The equation used to estimate these variables (WCHPMP or WCTPMP) is standard and may be found in any engineering handbook (22). The equation requires that the system pressure drop and the pump efficiency be estimated. These variables are calculated using quadratic models which relate the pressure drop, or the efficiency, to the fractional flow ratio. This ratio is the current system flowrate divided by the maximum possible

flowrate through the network. These empirical relationships are the same ones used in the full scale simulation and have coefficients which were determined using data provided by IBM. (Also note that the work of the hot water pumping network is not calculated by algorithms, as this work is usually small compared to the other electrical cost.)

The final set of computations in the pumping algorithm determine the fan work of the cooling tower. This work was found to be primarily a function of the tower load and its approach temperature. The quadratic model for the fan work is:

$$\begin{aligned}\text{Fan work} = {} & 4.96(QC1)-5.622(CTA)+3.904(QC1)(QC1) \\ & -0.409(QC1)(DTC)-15.677(QC1)(CTA) \\ & -2.933(DWB)(CTA)+14.805(CTA)+0.7315 \end{aligned} \quad (6)$$

where:

QC1 is the scaled cooling tower load.

DTC is the scaled temperature drop across the tower.

DWB is the scaled ambient wet bulb.

CTA is the scaled tower approach temperature.

Equation 6 has to be multiplied by 57.88 to rescale it to the appropriate range. Also, this equation is not given explicitly in the algorithm of Figure 5.8 instead
coefficients supplied in the array FCOF1. The values of the terms in this array are given in a data statement in Figure 5.4 (page 88).

One problem with quadratic models is that beyond the range of conditions where the coefficients were determined, the equations are not reliable. For this reason the value of WCTS is reset to 3 KWH if the computed value is negative or if the approach temperature is greater than 7°F (CTA > 0.5).

The pumping subroutine returns control of the computations to line 10 of the Mode 1.8 algorithm. The final task of this algorithm is to determine if any equipment constraints will be violated if the mode is implemented. The constraints that have to be satisfied are:

1. The computed compressor work has to be within the operating range of the machine.
2. There has to be enough cooling load to provide the reject heat required by the hot water loop (QCT < 0.0).
3. The tower set point cannot approach the ambient wet bulb temperature too closely.
4. The hot water supply temperature cannot exceed the safe operating temperature of the heat pump.

If any of these constraints are violated the algorithm sets the value of IVMOD [1] (for Mode 1.8) equal to zero and the calling routine will know that this mode should not be implemented.

This concludes the discussion of Mode 1.8. A list of variable names used in the algorithm is included in Table 5.7. In Table 5.8 is a set of typical input parameters which might be provided to the subroutine MODS. In all of the algorithms to follow, a set of sample calculations will be supplied to aid in the implementation of the quadratic models. These input variables should all be scaled as per Figure 5.4 (page 88). The calculated value of the COP for Mode 1.8 with the input parameters of Table 5.8 is 3.73708. The sample calculations for the pumping algorithms of Figure 5.8 are included in Table 5.9.

In Figure 5.9 is the Mode 1.9 algorithm. This algorithm differs from Mode 1.8 only in the values of the coefficients used in the COP model. The calculated value of the COP, using the input parameters of

TABLE 5.7

DEFINITION OF TERMS IN THE ALGORITHM FOR MODE 1.8

Term	Definition
COP	Estimated efficiency of the heat pump
QH, TSTX, THWR, DTH, DTC1	Scaled variables - (see Table 5.6, page 91)
Q3CH	Computed cooling load on heat pump
W3COMP	Computer heat pump compressor work
QCT	Computed cooling tower load (total load minus Q3CH)
TCTR	Temperature of water supplied by the cooling tower
THMAX	Maximum safe condensor exit temperature of the heat pump
PCOST	Pumping work
WCTS	Cooling tower fan work

TABLE 5.8

VALUES OF THE INDEPENDENT VARIABLES INPUT TO MODS AND USED IN THE SAMPLE CALCULATIONS

Variable	Magnitude	Scaled by
QCHUD	4.27×10^6	15×10^6
QHWUD	1.955×10^6	10×10^6
TCHSD	44.0	42.0
TSET	42.0	42.0
THWSD	100.0	90.0
DTCH	8.0	12.0
DTHOT	10.0	12.0
TWBOA	40.0	80.0
CRATIO	0.045/0.009	--

TABLE 5.9

VALUES OF VARIABLES CALCULATED BY THE PUMPING ALGORITHM OF FIGURE 5.8, USING THE INPUT PARAMETERS OF TABLE 5.8

Variable	Estimated Value Calculated from the Quadratic Model
DPCHP	2.917109
EFF1	0.31048
WCHPMP	4.36279
DPTWR	10.67507
EFF2	0.163307
WCTPMP	21.58351
WCTS	2.81495

```
C-----FREE COOLING MODE 1-9
C-----RM-2 AS A TOTAL HEAT RECOVERY DEVICE
C
15      CONTINUE
C----ESTIMATE THE COMPRESSOR WORK
C
        COP=1.949*QH+0.7388*TSETX-1.466*THWR-0.1627*DTH
        1   -1.062*(QH**2)+1.0136
C
        COP=3.0*COP
C
C
        Q2CH=(COP*QHWUD)/(1.0+COP)
        QCH=Q2CH
        W2COMP=(QHWUD-Q2CH)/3412.0
C
C-----COMPUTE THE COOLING TOWER CONTRIBUTION TO THE TOTAL COST
C
        IRET=2
        GOTO 1000
20      CONTINUE
C-----DETERMINE THE VALIDITY OF THIS MODE
C
        IVMOD(2)=1
        IF(W2COMP.GT.W2MAX)IVMOD(2)=0
        IF(W2COMP.LE.0.0)IVMOD(2)=0
        IF(QCT.LE.0.0)IVMOD(2)=0
        IF(TCTR.LT.(TWBOA+2.0))IVMOD(2)=0
        IF(THWDS.GT.THMAX)IVMOD(2)=0
C
C-----UPDATE THE COST
        IF(W2COMP.LT.W2MIN)W2COMP=W2MIN
        COST=WCTS+W2COMP+PCOST
C
C----RETURN THE ESTIMATE IN KWH
        CMOD(2)=COST
```

Figure 5.9. Algorithm for Mode 1.9.

Table 5.8, is 1.47377. The pumping subroutine algorithm for Mode 1.9 is the same as for Mode 1.8.

Free cooling with steam heat Mode 2. The computations required to evaluate Mode 2 are exactly the same as those used in Mode 1, except here the building's heating load is met by heating the hot water with low pressure steam.

In Figure 5.10 is the algorithm used in evaluating Mode 2. The algorithm simply sets the cooling tower supply temperature, TCTR, equal to the chilled water temperature, calculates the chilled water flow, sets the chilled water pumping work equal to zero (these pumps are not used in Mode 2), and then gives control to the pumping subroutine previously discussed in Mode 1. The subroutine is entered at line 1001.

When the subroutine returns control to the main algorithm, the steam requirement of the hot water network is calculated from the load and the latent heat of condensation of steam. The pounds-mass of steam needed is converted to equivalent kilowatts by multiplying by the cost ratio (CRATIO) of steam to electricity. The final step is to evaluate the constraints which cannot be violated if the mode is to be operational. The only constraint presently considered is the approach temperature constraint. If the requested chilled water temperature is less than two degrees above the ambient wet bulb, the mode is considered invalid and the value of IVMOD[2] is cleared to zero. Other constraints might be important but this one was the only one that ever occurred in the full scale simulation. The sample calculations have already been given in Table 5.9.

```
C-----ESTIMATE THE COST OF MODE 2-0
C-----FREE COOLING WITH STEAM HEAT
25      CONTINUE
C
        TCTR=TCHSD
        QCT=QCHUD
        WFCT=QCT/DTCH
        WCHPMP=0.0
        IRET=3
        TDXT=42.0
C----BRANCH TO PUMP EVALUATION SUBROUTINE
        GOTO 1001
C-----RETURN HERE FROM THE SUBROUTINE
30      CONTINUE
C----- COMPUTE STEAM CONSUMPTION
C
        STM=QHWUD/HSTM
C------DETERMINE VALIDITY OF THE MODE
        IVMOD(3)=1
        IF(TCTR.LT.(TWBOA+2.0))IVMOD(3)=0
C
        COST=WCTS+STM*CRATIO+PCOST
        CMOD(3)=COST
```

Figure 5.10. Algorithm for Mode 2.0.

Mechanical refrigeration with steam heat Mode 3. In Mode 3 the building's chilled water needs are provided by some combination of the system's three chillers. The hot water required by the building (if any) is supplied using low pressure steam.

The algorithms which evaluate the three all electric chiller submodes are essentially the same. The total electrical demand of each submode is first estimated using a simple quadratic model, then the hot water system's steam requirements are calculated, and finally the equipment constraints are tested to determine if the submode is operational. In the two steam chiller submodes (Modes 3.4 and 3.5), expressions are needed for estimating the steam required to drive the turbine of the chiller, but in other respects the algorithms are no different than the ones of the all electric chiller submodes.

For the sake of brevity and conciseness, the algorithm for Mode 3.2 will be discussed in detail and any differences in the algorithms of the other submodes will be pointed out in a brief discussion of each one.

Mode 3.2. In Mode 3.2 the large electric chiller (RM-2) is the only chiller running to meet the building's chilled water needs. The algorithm for evaluating the cost of this submode is given in Figure 5.11.

The algorithm for Mode 3.2 begins by first estimating the total electrical demand of the water system. All of the electricity consuming devices are lumped together into one simple quadratic expression which estimates this demand. The demand was found to be represented exceptionally well by a function whose independent variables are:

1. The chilled water load (QC).

```
C------ NEXT COMPUTE THE COST OF MODE 3-2
C       RM-2(1250 TON ELECTRIC CHILLER)
C
50     CONTINUE
C------FIRST COMPUTE THE TOTAL ELECTRIC DEMAND OF THE MODE
C
       WCTS=-1.219*QC-0.6657*TCHS+1.068*TDWB+3.020*(QC**2)+0.4173
       WCTS=771.66*WCTS
C
C------NOW ADD ON THE COST OF STEAM HEAT
C
       STM=QHWUD/HSTM
C
C
C------COMPUTE TOTAL ESTIMATE OF COST
       COST=WCTS+STM*CRATIO
C
C------RETURN COST IN CMOD(5)
       CMOD(5)=COST
C
C-----DETERMINE IF THE COMPRESSOR CAN MEET THE LOAD
C      CALCULATE THE COP
C----DETERMINE THE CONDENSOR INLET TEMPERATURE
       T2CDI=TWBOA+5.0
C----MAKE SURE TICDI IS NOT LESS THAN 55 THEN SCALE
       IF(T2CDI.LT.55.0)T2CDI=55.0
       T2CDI=T2CDI/80.0
C
       COP=2.280*QC+1.1359*TCHS-1.8503*T2CDI-0.0453*DTC
       1  -2.7623*(QC**2)+1.5271
C
       COP=3.0*COP
C
C      COMPUTE THE COMPRESSOR WORK
       W2COMP=QCHUD/(COP*3412.0)
C
C
       IVMOD(5)=1
       IF(W2COMP.GT.W2MAX)IVMOD(5)=0
       IF(W2COMP.LE.0.0)IVMOD(5)=0
```

Figure 5.11. Algorithm for Mode 3.2.

2. The chilled water supply temperature (TCHS).

3. The ambient wet bulb temperature (DWB).

The total energy of the submode is calculated as the sum of the total electrical demand (WCTS) and the converted steam demand (STMC) of the hot water loop.

After determining the cost, the equipment constraints are evaluated. In Mode 3 the only operating constraint considered is whether the chiller's estimated compressor work is within the operating range of the device. The compressor work is estimated using an efficiency equation similar to the COP expression of Mode 1. This equation differs in that the cooling load on the chiller is known and the heat rejected by the chiller is of no concern. The mode is invalid if the estimated compressor work constraint is violated.

The two quadratic models used in Mode 3.2 compute the systems electrical demand (WCTS) and the chiller efficiency (COP). The computed values of these variables using the set of input conditions of Table 5.8 (page 104) are 271.5735 and 5.5203, respectively.

Mode 3.1. In Mode 3.1 the small electric chiller (RM-3) is running. The algorithm for this submode is provided in Figure 5.12. The differences between the 3.1 and the 3.2 algorithms are minor. The coefficients in the WCTS and the COP equations are different and the submode has one additional constraint to be evaluated. This constraint is: If the system load is far greater than the nominal capacity of RM-3, the mode is set invalid. Since the COP equation for Mode 3.1 is scaled for a load of 3.0 million, loads that are much greater than this will cause the quadratic equation to give unreliable answers.

```
C-----NOTE MODE 4-X IS EQUIVALENT TO MODE 3-X EXCEPT
C     THAT PAY HEAT IS REQUIRED IN THE FORM OF STEAM
C-----START WITH MODE 3-1
C----- RM-3(250 TON CHILLER)
C
40     CONTINUE
C-----ESTIMATE THE TOTAL COST OF THE SUBMODE
C
       WCTS=0.52*QC-0.37*TCHS+1.13*TDWB+21.29*(QC**2)
       WCTS=155.92*WCTS
C
C
C------NOW COMPUTE THE STEAM USAGE FOR HEATING
       STM=QHWUD/HSTM
C
C------FINALLY COMPUTE THE TOTAL EQUIVALENT ELECTRICAL COST
C
       COST=WCTS+STM*CRATIO
C-----NEXT DETERMINE IF THE COMPRESSOR CAN MEET THE LOAD
C
C-----ESTIMATE THE COP
C----- ASSUME CONDENSOR INLET TEMPERATURE IS 5.0 DEGREES
C     ABOVE THE AMBIENT WETBULB UNLESS THE WETBULB IS
C     LESS THAN 55.0 DEGREES F.
C-----TICDI IS THE INLET CONDENSOR TEMPERATURE
       TICDI=TWBOA+5.0
C-----MAKE SURE TICDI IS NOT LESS THAN 55 THEN SCALE IT
       IF(TICDI.LT.55.0)TICDI=55.0
       TICDI=TICDI/80.0
       COP=1.152*QC*(15.0/3.0)+0.9447*TCHS-0.1162*DTC
       1   -1.495*TICDI-0.8394*((QC*(15.0/3.0))**2)+1.6599
       COP=3.0*COP
C
C-----ESTIMATE THE COMPRESSOR WORK
       QR=QCHUD+(QCHUD/COP)
       W3COMP=(QR-QCHUD)/(3412.0)
C
C-----DETERMINE THE VALIDITY OF THIS SUBMODE
       IVMOD(4)=1
       IF(W3COMP.GT.W3MAX)IVMOD(4)=0
       IF(W3COMP.LE.0.0)IVMOD(4)=0
       IF(QCHUD.GT.5.0E6)IVMOD(4)=0
       CMOD(4)=COST
```

Figure 5.12. Algorithm for Mode 3.1.

The estimated values of WCTS and COP, using the same inputs as for Mode 3.2, are 353.035 and 4.4479.

Mode 3.3. The other all electric chiller mode is Mode 3.3. In this mode the two chillers are running in parallel. The algorithm for this mode is presented in Figure 5.13. This algorithm is very similar to those of the other two submodes. The constraints are evaluated by estimating the compressor work of the large chiller. The algorithm assumes that RM-2 is meeting about 70% of the total load. This assumed load is then used to determine the compressor work from the COP equation used first in Mode 3.2. If the estimated work exceeds the maximum capacity of RM-2, the submode is set invalid. Note that if system loads do exceed the capacity of this mode, the system design loads were probably too small.

For input conditions supplied in Table 5.8, page 97, the calculated value of WCTS is 287.5334. The COP was previously tested in Mode 3.2.

Mode 3.4. Mode 3.4 is the first mode with the steam chiller running. The electrical demand of this mode is estimated using a quadratic model. The steam demand is estimated from the quadratic efficiency equation for the steam chiller. The procedure is to estimate the COP, then calculate the compressor work. The steam demand is related to compressor work by a linear equation whose coefficients were determined from data provided by IBM. The equation for conversion is (see Figure 5.14):

$$STMC = 891.42 + 0.004973(W1COMP) \qquad (7)$$

where:

STMC is the pounds per hour of steam.

```
C
C------NEXT COMPUTE THE COST OF MODE 3-3
C       RM2(1250 TON ELECTRIC CHILLER) IN PARALLEL WITH
C       RM3(250 TON ELECTRIC CHILLER)
C
60      CONTINUE
C
C-------COMPUTE THE ELECTRICAL DEMAND OF THE MODE
C
        WCTS=0.3269*QC-0.6981*TCHS+1.1681*TDWB-0.0622*DTC
        1    +1.2111*(QC**2)+0.1945
        WCTS=691.03*WCTS
C
C------NEXT COMPUTE THE STEAM CONSUMPTION FOR HEATING
C
        STM=QHWUD/HSTM
C
C-----COMBINE TO GET THE TOTAL COST
        COST=WCTS+STM*CRATIO

C-----RETURN COST IN CMOD ARRAY
        CMOD(6)=COST
C-----NOW DETERMINE IF THE MODE IS VALID
C-----  ASSUME RM-2 IS MEETING 71.5 % OF THE SYSTEM LOAD
        Q2CH=(0.715*QCHUD)/(15.0E6)
C-----DETERMINE IF THE COMPRESSOR CAN MEET THE LOAD
C       CALCULATE THE COP
        T2CDI=TWBOA+5.0
        IF(T2CDI.LT.55.0)T2CDI=55.0
        T2CDI=T2CDI/80.0
C
        COP=2.280*Q2CH+1.1359*TCHS-1.8503*T2CDI-0.0453*DTC
        1  -2.7623*(Q2CH**2)+1.5271
C
        COP=3.0*COP
C
C       COMPUTE THE COMPRESSOR WORK
        W2COMP=Q2CH*15.0E6/(COP*3412.0)
C
C
C-----SET THE MODE AS VALID
        IVMOD(6)=1
C
        IF(W2COMP.LT.5.0)IVMOD(6)=0
```

Figure 5.13. Algorithm for Mode 3.3.

```
C------NEXT COMPUTE THE COST OF MODE 3-4
C       RM1(1250 TON STEAM CHILLER)
70      CONTINUE
C------FIRST ESTIMATE THE ELECTRICAL CONSUMPTION FOR THIS MODE
C
        WCTS=-0.4155*QC-0.1433*TCHS-0.7510*TDWB-0.2903*DTC
        1     +1.7051*(QC**2)+1.6131
        WCTS=165.9*WCTS
C
C------NEXT ESTIMATE THE STEAM CONSUMPTION IN THE STEAM CHILLER
C
        STMC=-0.9116*QC-0.7054*TCHS+1.3037*TDWB+2.7368*(QC**2)+.2742
        STMC=10686*STMC
C       ESTIMATE THE COP
C
        T1CDI=TWBOA+5.0
        IF(T1CDI.LT.55.0)T1CDI=55.0
        T1CDI=T1CDI/80.0
        COP=1.6282*QC+1.9228*TCHS
        1  -18.3418*T1CDI-0.06732*DTC
        2  -2.0625*(QC**2)+9.2097*(T1CDI**2)+8.3035
C
        COP=3.0*COP
C
        W1COMP=QCHUD/COP
        STMC=891.42+0.004973*W1COMP
        W1COMP=W1COMP/3412.0
C
C------FINALLY ESTIMATE THE STEAM USED FOR HEATING
        STMH=QHWUD/HSTM
C
C------SUM UP STEAM USED AND CONVERT TO EVUIVALENT ELECTRICITY
C
        STM=(STMC+STMH)*CRATIO
C
C------COMPUTE THE TOTAL COST OF THE MODE AND STORE IN CMOD(7)
        COST=WCTS+STM
        CMOD(7)=COST
C-------SET THE MODE AS VALID
C       DETERMINE IF THE COMPRESSOR CAN MEET THE LOAD
        IVMOD(7)=1
        IF(W1COMP.GT.W1MAX)IVMOD(7)=0
        IF(W1COMP.LE.0.0)IVMOD(7)=0
```

Figure 5.14. Algorithm for Mode 3.4.

W1COMP is the estimated compressor work in BTU's per hour.

The total energy cost is again the electrical demand plus the converted steam cost. The converted steam cost is the sum of the steam required in the hot water network and that required by the steam chiller.

The calculated values of WCTS and the COP are 128.257 and 6.937 if the inputs from Table 5.8 (page 97) are used.

Mode 3.5. The final Mode 3 submode modeled is Mode 3.5. In this submode the steam chiller and the small electric chiller are running in parallel. In the algorithm for this mode, given in Figure 5.15, the steam rate of the chiller and the electrical demand of the water system are estimated directly using quadratic models. The direct model of the steam chiller gave better results with fewer computations than would have been required if the COP equation had been used. (The COP equation would have required calculating an estimate of the portion of the total load met by RM-1.)

The only equipment constraint tested in the algorithm is whether the system load is too low for the safe operation of the submode. The submode is operated optimally when the small chiller is based loaded; for light loads this means the steam chiller would be virtually shut off. This submode is obviously not economical to operate under light loads. Thus if the system load approaches the nominal capacity of RM-3, the mode is set invalid.

The estimated value of WCTS and STMC for this submode should be 298.8788 and 949.2842.

Mechanical refrigeration with heat pump Mode 5. Mode 5 is very similar to Mode 1.8, but instead of the cooling tower being in parallel

```
C--------NEXT COMPUTE THE COST OF MODE 3-5
C         RM1(1250 TON STEAM CHILLER) IN PARALLEL WITH
C         RM3(250 TON ELECTRIC CHILLER)
C
C
80      CONTINUE
C
C--------COMPUTE THE ELECTRICAL DEMAND OF THE MODE
C
        WCTS=0.55*QC+0.45*TCHS+0.15*TDWB+0.05*(QC**2)
        WCTS=406.5*WCTS
C
C--------NOW COMPUTE THE STEAM USAGE FOR THE STEAM CHILLER
C
        STMC=0.5234*QC-1.0139*TCHS+1.9278*TDWB+0.0251*DTC
        1   +1.5978*(QC**2)-0.4152
        STMC=6620.83*STMC
C
C-------NEXT COMPUTE THE STEAM REQUIRED FOR HEATING
C
        STMH=QHWUD/HSTM
C
C-------SUM UP THE STEAM USED AND CONVERT TO EQUIVALENT ELECTRICAL UNIT
S
C
        STM=(STMC+STMH)*CRATIO
C
C------COMPUTE THE TOTAL COST ESTIMATE AND STORE IN CMOD(8)
        COST=WCTS+STM
        CMOD(8)=COST
C------DETERMINE IF MODE IS VALID
C
        IVMOD(8)=1
        IF(QCHUD.LT.4.0E6)IVMOD(8)=0
C
```

Figure 5.15. Algorithm for Mode 3.5.

with the heat pump's evaporator, RM-1 or RM-2 is connected in parallel with it. In both the submodes studied RM-3 is always the system's total heat recovery device. The algorithms used to evaluate the submodes are similar to ones which have been previously discussed. The algorithm for Mode 5.11 follows the pattern of the Mode 1.8 algorithm. The calculations for Mode 5.12 are similar to those of Mode 3.4.

Mode 5.11. Mode 5.11 is an all electric heat recovery mode. The algorithm is given in Figure 5.16. The algorithm begins by estimating the heat pump's efficiency from the known system hot water load. The heating load and the estimated COP are then used to estimate the heat pump's evaporator load. The equation for the COP was discussed previously in the discussion of Mode 1.8.

After determining RM-3's evaporator load (QsCH), RM-2's is calculated by substracting Q3CH from the total load QCHUD. The compressor work of RM-2 is estimated using the equation of its efficiency: This COP expression was described in the Mode 3.2 algorithm. From the evaporator load and the compressor work, the heat which must be rejected by the cooling tower is calculated. The flow through the cooling tower network is known from optimization studies of Chapter V. The cooling water flowrate, the chilled water flowrate, the cooling tower load, and the tower supply temperature are now known. (The tower set point is given by a procedural optimization on page 67.) The algorithm now calls the pumping subroutine to determine the cooling tower fan work and the pumping energy required in the chilled and cooling tower water pumping networks. This subroutine, discussed in Mode 1, is given in Figure 5.8 on page 98. When the subroutine returns control to the Mode 5.11 algorithm, at line number 95

```
C-----NEXT COMPUTE THE COST OF MODE 5-11
C     RM2 AS A CHILLER
C     RM3 AS A TOTAL HEAT RECOVERY DEVICE
C
90     CONTINUE
C----FIRST ESTIMATE THE HEAT PUMP'S COP
C
       COP=2.307*QH*(10.0/3.0)+0.6002*TSETX-1.108*THWR
       1 -0.0670*DTH-0.0177*DTC1-1.269*((QH*(10.0/3.0))**2)
       2 +0.85959
C
       COP=3.0*COP
C
C----- NOW ESTIMATE THE EVAPORATOR LOAD ON THE HEAT PUMP
C
       Q3CH=(COP*QHWUD)/(1.0+COP)
C
C----- THEN CALCULATE THE COMPRESSOR WORK IN KWH
C
       W3COMP=(QHWUD-Q3CH)/3412.0
C
C---- ESTIMATE THE EVAPORATOR LOAD ON RM-2
C
       Q2CH=QCHUD-Q3CH
C
C----- AND SCALE IT
C
       Q2CH=Q2CH/15.0E6
C----- NOW ESTIMATE THE COMPRESSOR WORK OF RM-2
C      CALCULATE THE COP
C
       T2CDI=TWBOA+5.0
       IF(T2CDI.LT.55.0)T2CDI=55.0
       T2CDI=T2CDI/80.0
C
       COP=2.280*Q2CH+1.1359*TCHS-1.8503*T2CDI-0.0453*DTC
       1  -2.7623*(Q2CH**2)+1.5271
C
       COP=3.0*COP
C
       W2COMP=(Q2CH*15.0E6)/(COP*3412.0)
```

Figure 5.16. Algorithm for Mode 5.11.

```
C-----NOW CALCULATE THE PUMP AND COOLING TOWER WORK
C
C----- SET THE COOLING TOWER FLOW RATE AS PER THE PROCEDURAL
C     OPTIMIZATION OF CHAPTER FOUR
C
       WFCT=971000
C
C------NEXT CALCULATE THE HEAT REJECTED TO THE COOLING TOWER
C----- IT IS THE COOLING LOAD PLUS THE COMPRESSOR WORK
C
       QCT=Q2CH+W2COMP*3412.0
C
C-------NOW THE CHILLED WATER FLOW RATE
C
       WFCH=QCHUD/DTCH
C
C-----COOLING TOWER SET POINT
C
       TCTR=TWBOA+5.0
C
       WFLOT=WFCH
C
       IRET=4
C
C----- NOW GO TO THE PUMPING SUBROUTINE
       GOTO 1020
95     CONTINUE
C
C------ EVALUATE THE EQUIPMENT CONSTRAINTS
C
       IVMOD(9)=1
       IF(W3COMP.GT.W3MAX)IVMOD(9)=0
       IF(W2COMP.LE.0.0)IVMOD(9)=0
       IF(W3COMP.LE.0.0)IVMOD(9)=0
       IF(THWSD.GT.THMAX)IVMOD(9)=0
C
C------ NOW COMPUTE THE ESTIMATED ENERGY COST
C
       IF(W2COMP.LT.W2MIN)W2COMP=W2MIN
       IF(W3COMP.LT.W3MIN)W3COMP=W2MIN
       COST=WCTS+PCOST+W2COMP+W3COMP
       CMOD(9)=COST
```

Figure 5.16. (Continued)

in Figure 5.16, the equipment constraints are evaluated to determine if the mode is operational. The constraints considered are:

1. Whether the calculated compressor work of the heat pump is within the operating range of RM-3.
2. If the requested hot water supply temperature is below the maximum temperature the heat pump can safely supply.
3. If the chiller can meet the load on its evaporator.

The value of IVMOD[9] is set to zero if any of these equipment constraints are violated.

The total energy cost for the submode is computed as the sum of the work in the two compressors, the work in the two pumping loops, and the cooling tower fan work. This cost is returned in CMOD[9].

Because all of the quadratic models used in this algorithm have been discussed in other modes, no sample calculations are needed for Mode 5.11.

Mode 5.12. In this submode, RM-1 is the system chiller. The algorithm for evaluating Mode 5.12 is given in Figure 5.17. The steam requirements are determined by estimating the chiller's evaporator load. The procedure for estimating this load is the same one used in Mode 5.11. The COP equation is the same expression used in the Mode 3.4 algorithm. The electrical demand of the submode is estimated by a single equation which uses both the heating and cooling loads, along with the other system variables, to evaluate this demand. The constraints are the same as in Mode 5.11.

The value of WCTS which the algorithm calculates using the inputs of Table 5.8 (page 103) is 216.16.

```
C------NEXT ESTIMATE THE COST OF MODE 5-12
C       RM1 AS A CHILLER AND
C       RM3 AS A TOTAL HEAT RECOVERY DEVICE
C
100     CONTINUE
C
C------FIRST ESTIMATE THE ELECTRICAL DEMAND
C
        WCTS=-0.6145*QC+0.2561*QH*(10.0/3.0)-0.3401*TCHS
        1    +0.5677*THWR-0.3478*TDWB-0.0528*DTC+1.1007*(QC**2)
        2    +0.27346*((QH*(10.0/3.0))**2)+0.7902
C
        WCTS=234.0*WCTS
C
C------NEXT ESTIMATE THE SMALL CHILLER COP
C
        COP=2.307*QH*(10.0/3.0)+0.6002*TSETX-1.108*THWR
        1 -0.0670*DTH-0.0177*DTC1-1.269*((QH*(10.0/3.0))**2)
        2 +0.85959
C
        COP=3.0*COP
C
C-----ESTIMATE THE HEAT PUMP COMPRESSOR WORK FOR RM3
C
        Q3CH=(COP*QHWUD)/(1.0+COP)
        QCH=Q3CH
        W3COMP=(QHWUD-QCH)/3412.0
C
C------NOW ESTIMATE THE STEAM USAGE IN THE STEAM CHILLER
C-----  CALCULATE THE TOTAL CHILLED WATER FLOW RATE
C
        FLOWT=QCHUD/DTCH
C
C-----NOW CALCULATE THE CHILLED WATER FLOW THROUGH RM3
C
        FLOW3=Q3CH/(DTC1*12.0)
C------FINALLY THE FLOW THROUGH RM1
C
        FLOW1=FLOWT-FLOW3
C------CALCULATE THE LOAD ON THE STEAM CHILLER AS THE TOTAL
C       CHILLED WATER LOAD MINUS THE LOAD ON RM3
C
        Q1CH=QCHUD-Q3CH
```

Figure 5.17. Algorithm for Mode 5.12.

```
C-----NEXT ESTIMATE THE TEMPERATURE DROP ACROSS RM1
C
       DT1=Q1CH/FLOW1
C
C----- THEN THE REQUIRED STEAM CHILLER CHILLED WATER TEMPERATURE
C
       T1CH=TCHSD+DTCH-DT1
C
C-------NOW COMPUTE THE SCALED COOLING TOWER WATER TEMPERATURE
C
       T1CDI=TWBOA+5.0
       IF(T1CDI.LT.55.0)T1CDI=55.0
       T1CDI=T1CDI/80.0
C
C-----NOW CALCULATE THE COP OF THE STEAM CHILLER
C
       COP=1.6282*(Q1CH/15.0E6)+1.9228*(T1CH/42.0)
       1  -18.3418*T1CDI-0.06732*(DT1/12.0)
       2  -2.0625*(((Q1CH/15.0E6)**2))+9.2097*(T1CDI**2)+8.3035
C
       COP=3.0*COP
C
C------NEXT ESTIMATE THE COMPRESSOR WORK OF RM1
C
       W1COMP=Q1CH/COP
C
C------CONVERT THE COMPRESSOR WORK TO STEAM CONSUMPTION
C
       STMC=891.42+0.004973*W1COMP
       W1COMP=W1COMP/3412.0
C
C------NOW DETERMINE IF THE MODE IS VALID
C
       IVMOD(10)=1
       IF(W3COMP.GT.W3MAX)IVMOD(10)=0
       IF(W3COMP.LE.0.0)IVMOD(10)=0
       IF(W1COMP.LE.0.0)IVMOD(10)=0
       IF(THWSD.GT.THMAX)IVMOD(10)=0
C
C-----COMPUTE THE TOTAL ESTIMATE OF THE COST
C
       COST=WCTS+STMC*CRATIO
       CMOD(10)=COST
```

Figure 5.17. (Continued)

Partial heat recovery Mode 6. The final mode of operation modeled is Mode 6. In this mode one of the two heat pumps is supplying all of the building's hot and chilled water. An equipment schematic is provided in Figure 5.18. From the Figure one sees that the cooling tower is used to reject any of the heat pump's excess heat. The hot water temperature controller manipulates the tower bypass valve to achieve the desired hot water temperature. The algorithms for Mode 6 have to determine if the system loads are such that if this mode is implemented the bypass valve will neither be fully open or fully closed. Under those circumstances the submode could not meet the needs of the hot water users.

The algorithm for Mode 6.1 is given in Figure 5.19. The electrical demand for the mode is estimated by a simple quadratic expression. The equipment constraints are evaluated by first determining the heat rejected by the heat pump with the condensor inlet temperature equal to the temperature returned by the hot water users. This corresponds to the condition where the cooling load is not great enough and the bypass is fully open (the cooling tower is getting minimal flow through it). The COP equation for a heat pump is used to determine the chiller efficiency. The reject heat is then calculated using the known cooling load and the estimated COP. This heat is QR in Figure 5.19.

The capacity of the hot water users and the cooling tower to reject heat is next estimated. The flowrate in the network is determined by the hot water users. The total capacity is determined by calculating the heat rejected by the tower in cooling the return hot water to a few degrees above the ambient wet bulb. If the cooling tower capacity (QCT) plus the hot water load (QHWUD) is less than QR, the mode is considered invalid. This is not actually the case because the efficiency of the chiller should

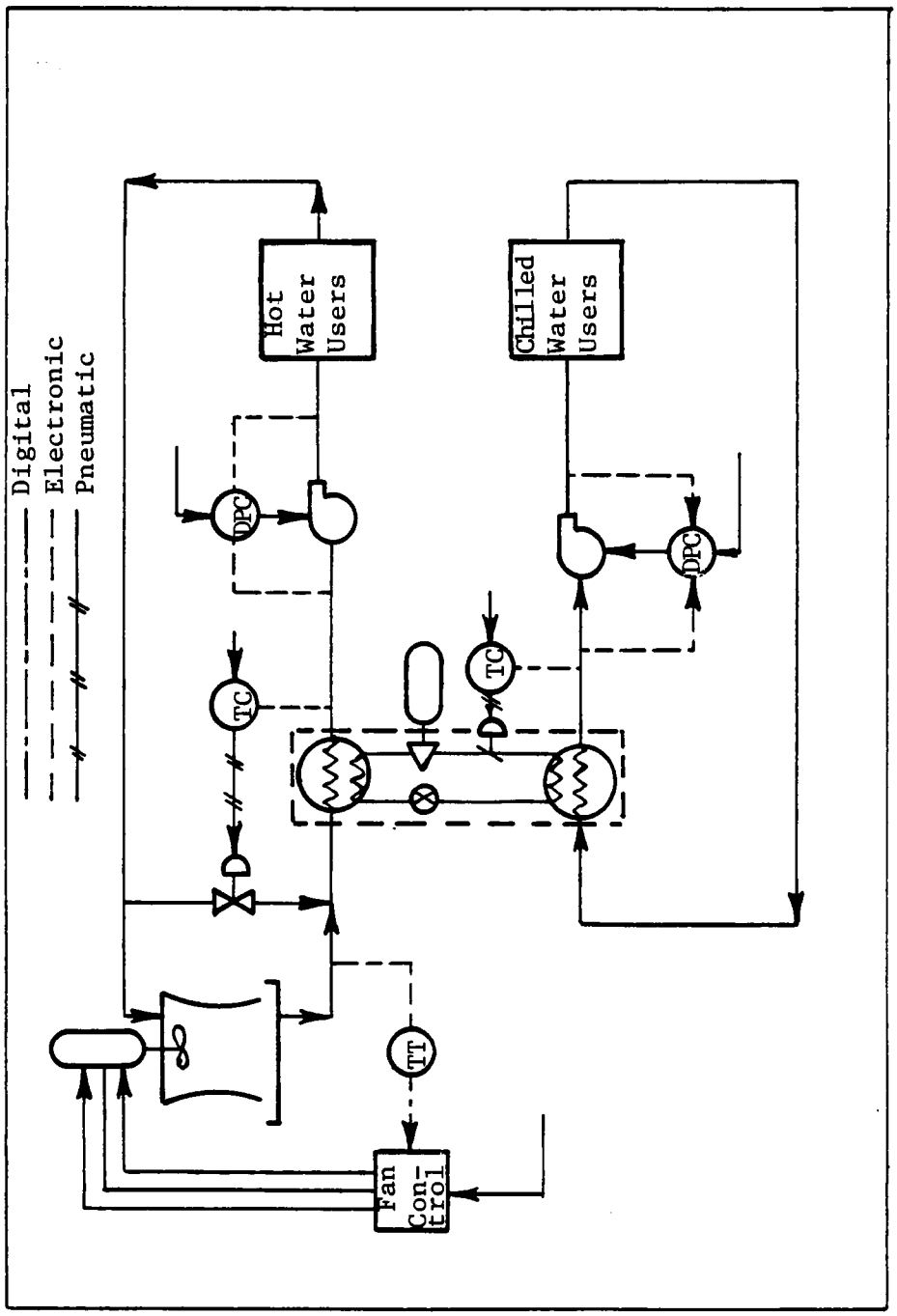


Figure 5.18. Partial heat recovery Mode 6. This mode has two submodes. In 6.1 the small chiller is on, in Mode 6.2 the large chiller is on.

```
C------NEXT ESTIMATE THE COST OF MODE 6-1
C
C       PARTIAL HEAT RECOVERY WITH RM3
C
110     CONTINUE
C
C------FIRST ESTIMATE THE ELECTRICAL DEMAND
C
        WCTS=0.18*QC*(15.0/3.0)+0.46*QH*(10.0/3.0)-0.39*TCHS
        1    +0.95*(THWSD/90.0)+0.99*((QC*(15.0/3.0))**2)
        2    -0.38*QC*(15.0/3.0)*QH*(10.0/3.0)
C
        WCTS=132.0*WCTS
C
C------NEXT ESTIMATE IF MODE IS VALID
C
C------FIRST ESTIMATE THE HEAT REJECTED BY RM3
C       FIRST GET A ROUGH ESTIMATE OF THE COP
C
C       USE THE HOT WATER RETURN TEMPERTURE AS STARTING
C       INLET TEMPERTATURE TO CONDENSOR
C
        COP=2.307*QH*(10.0/3.0)+0.6002*TCHS-1.108*THWR
        1 -0.0670*DTH-0.0177*DTC-1.269*((QH*(10.0/3.0))**2)
        2 +0.85959
C
        COP=3.0*COP
        W3COMP=QCHUD/COP
        QR=QCHUD+W3COMP
        W3COMP=W3COMP/3412.0
C
```

Figure 5.19. Algorithm for Mode 6.1.

```
C ------NEXT CALCULATE THE COOLING CAPACITY OF THE COOLING TOWER
C         HOT WATER USERS
C
C        CALCULATE THE HOT WATER FLOW
C
       FHW=QHWUD/DTHOT
C
C ----- TOTAL CAPACITY IS QHWUD +QCT(COOLING TOWER HEAT REJECTION)
C
C ----- NOTE THWSD-DTHOT IS THE TEMPERATURE OF THE WATER RETURNING
C      FROM THE HOT WATER USERS AND THAT TWBOA+6.0 IS THE SET POINT
C      OF THE COOLING TOWER
C
       QCT=FHW*(THWSD-DTHOT-TWBOA-6.0)
       QCAP=QHWUD+QCT
C------NOW DETERMINE IF THE MODE IS VALID
C------IF RM3 REJECTS TOO MUCH HEAT OR THE HEAT LOAD
C------IS GREATER THAN THE HEAT PUMP COOLING TOWER COMBINATION
C
       IVMOD(11)=1
       IF(QCAP.LT.QR)IVMOD(11)=0
       IF(W3COMP.GT.W3MAX)IVMOD(11)=0
       IF(W3COMP.LT.W3MIN)IVMOD(11)=0
       IF(QHWUD.GT.QR)IVMOD(11)=0
       IF(THWSD.GT.THMAX)IVMOD(11)=0
C
C-----FINALLY RETURN THE ESTIMATE OF THE TOTAL COST
C
       COST=WCTS
       CMOD(11)=COST
```

Figure 5.19. (Continued)

be evaluated at the condensor inlet temperature with the bypass fully closed to get a more accurate estimate of the reject heat. But for all the studies conducted using the maximum heat, it was always sufficiently accurate that the mode was not set valid when it was invalid or visa versa. The equipment constraints that are tested to determine if a submode is valid are:

1. The compressor work has to be within the operating range of the compressor.
2. The reject heat QR has to be greater than the total hot water load QHWUD.
3. The total capacity of the hot water plus cooling tower network has to be greater than QR.
4. The set point temperature for the hot water loop has to be less than the maximum safe temperature of the heat pump.

The algorithm for Mode 6.2 is given in Figure 5.20. The computations for this submode are equivalent to those of Mode 6.1. The estimated electrical demand (WCTS) of Mode 6.1 is 377.2494 and that of Mode 6.2 is 462.5325. The COP equations used in these algorithms were taken from Modes 1.9 and 1.9, respectively, therefore sample calculations are not provided. As always the input variables used in these sample calculations come from Table 5.8 (page 103) and scaling should be done as per Figure 5.4 (page 87).

Concluding Remarks

The algorithms described in this chapter should not be expected to predict the actual operating behavior of the building's mechanical equipment closely. Instead they suggest the correct form of the algorithms.

```
C------AND FINALLY COMPUTE THE COST OF MODE 6-2
C       RM2 AS A PARTIAL HEAT RECOVERY DEVICE
C
120     CONTINUE
C
C------FIRST ESTIMATE THE ELECTRICAL CONSUMPTION
C
        WCTS=0.79*QC+0.28*QH-0.93*TCHS+1.34*(THWSD/90.0)+1.23*(QC**2)
        1   +0.01*QC*QH
C
        WCTS=517.0*WCTS
C
C------NEXT ESTIMATE IF MODE IS VALID
C
C-----ESTIMATE THE COP TO BE 3.5
C      AND THEN CALCULATE THE REJECT HEAT FROM THE COOLING LOAD
C
        QR=QCHUD +(QCHUD/3.5)
C
C      CALCULATE THE HOT WATER FLOW THROUGH THE COND.
C
        FHW=QHWUD/DTHOT
C
C----- THEN APPROXIMATE THE CONDENSOR INLET TEMPERATURE
C       AS INLET = EXIT (KNOWN) - DELTA T
C       (DELTA T IS THE ESTIMATED REJECT HEAT DIVIDED BY THE FLOW)
C
        T2CDI=(THWSD-(QR/FHW))/80.0
C
        COP=1.949*QH+0.7388*TCHS-1.466*T2CDI-0.1627*DTH
        1   -1.062*(QH**2)+1.0136
C
        COP=3.0*COP
        W2COMP=QCHUD/COP
        QR=QCHUD+W2COMP
        W2COMP=QCHUD/(COP*3412.0)
C
```

Figure 5.20. Algorithm for Mode 6.2.

```
C ------NEXT CALCULATE THE COOLING CAPACITY OF THE COOLING TOWER
C         HOT WATER USERS
C
C        CALCULATE THE HOT WATER FLOW
C
       FHW=QHWUD/DTHOT
C
C ----- TOTAL CAPACITY IS QHWUD +QCT(COOLING TOWER HEAT REJECTION)
C
       QCT=FHW*(THWSD-DTHOT-TWBOA-6.0)
       QCAP=QHWUD+QCT
C------NOW DETERMINE IF THE MODE IS VALID
C------IF RM3 REJECTS TOO MUCH HEAT OR THE HEAT LOAD
C------IS GREATER THAN THE HEAT PUMP COOLING TOWER COMBINATION
C
       IVMOD(12)=1
       IF(W2COMP.GT.W2MAX)IVMOD(12)=0
       IF(W2COMP.LT.W2MIN)IVMOD(12)=0
       IF(QCAP.LT.QR)IVMOD(12)=0
       IF(QHWUD.GT.QR)IVMOD(12)=0
       IF(THWSD.GT.THMAX)IVMOD(12)=0
C
C-----NOW RETURN THE COST
C
       COST=WCTS
       CMOD(12)=WCTS
```

Figure 5.20. (continued)

It is expected that once operating experience with the actual system is obtained, the algorithms can be easily updated.

The algorithms should be viewed in two ways:

1. In their present form they provide a mechanism for evaluating the relative merits of the system configurations over a wide range of operating conditions.
2. The algorithms provide the correct structure of the computations needed to evaluate the complex system which is available in 590.

The task of updating the algorithms should not be undertaken lightly. A good statistical design is needed to determine the correct values of the coefficients in the quadratic models. The preferred approach to this updating would be to gather partial loading data for the chillers, pumps, and other system components and then to update the adjustable parameters in the full scale simulation. The simulation is an excellent tool for studying large systems. The quadratic models can then be refit using the updated simulation results.

In the next chapter some examples of the kinds of studies which can be performed to evaluate the operation of the building will be discussed. These studies provide an indication of the potential energy savings that can be achieved by the development of a comprehensive understanding of the system through a steady state simulator. These studies no doubt represent just a small subset of the studies which are possible.

CHAPTER VI

DAILY OPERATION OF THE BUILDING

After making all the preliminary design and economical analysis concerning the type of heat reclaim system which is applicable to a specific building, some consideration must be given to the operational aspects of the system. With the mechanical equipment available today, the selected system may have a high degree of operational flexibility. If the higher initial capital outlay for such a system is to be worthwhile, there must be a comprehensive understanding of the various operating possibilities and their relative merits. In the water systems of 590 there are at least 13 distinct operating configurations which are able to meet the heating and cooling needs of the building. Under some outside and loading conditions, all 13 might be feasible simultaneously.

In Chapter V, a set of algorithms was discussed which would estimate the operating range of cost of system configurations for a wide range of operating conditions. The purpose of this chapter is to illustrate the effectiveness of the algorithms in selecting the optimal mode of operation of the water systems during a typical day of operation.

The hypothetical day is based upon a set of heating and cooling load profiles which might occur during the course of a day. The load profiles are for a day where the outside air temperature never rises above 50°F. In Figure 6.1 are plotted the load profiles. There are two heating loads and a single cooling load. The purpose of the two heating loads is to show the effect of a high and a low heating load profile upon the optimal operating path for a typical cooling load profile.

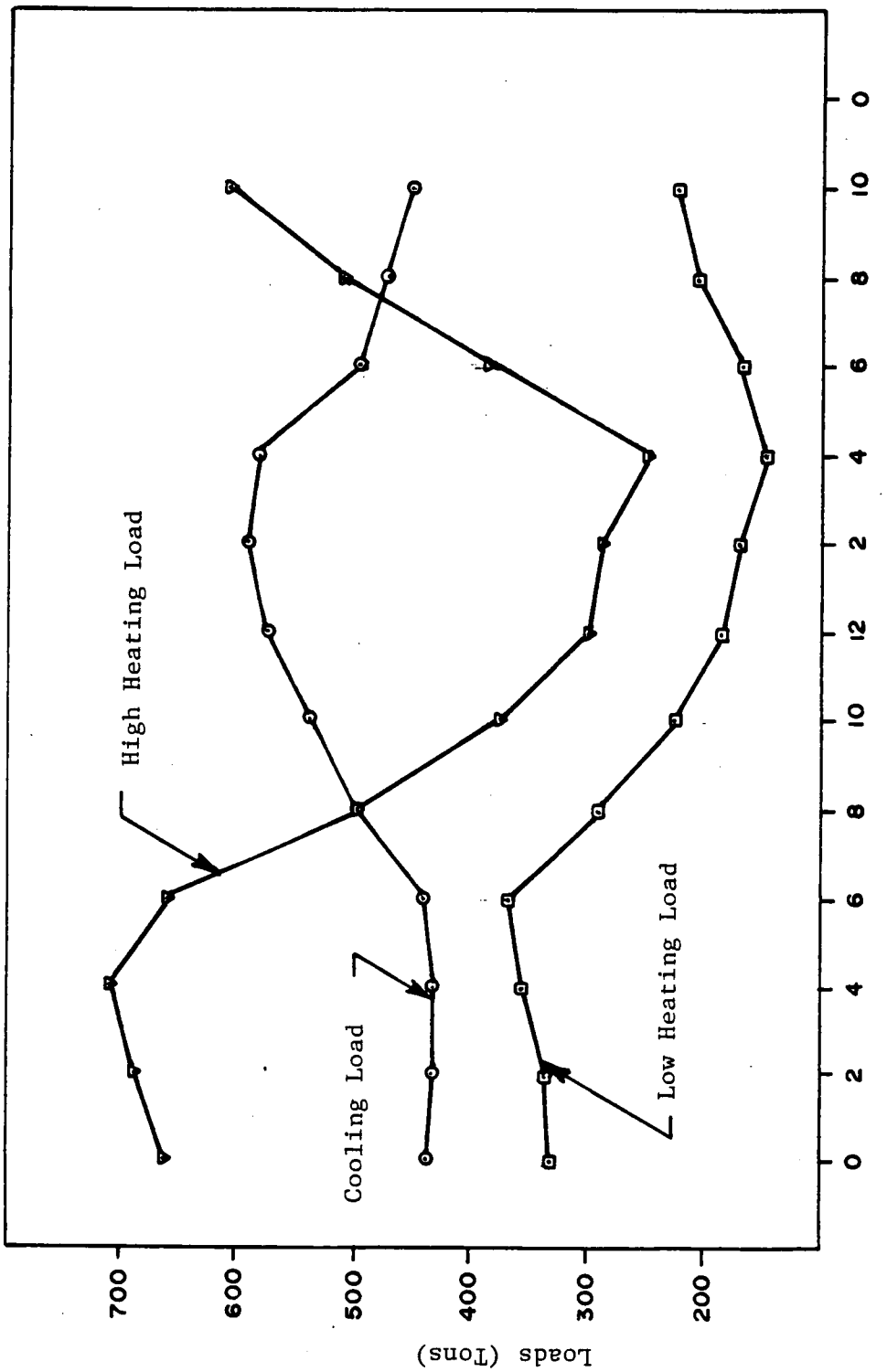


Figure 6.1. Hypothetical heating and cooling loads on the primary water system.

In Figure 6.2 are the cost curves for the first study of the system. The curves plotted on this figure are for the high heating load. As indicated by the figure, nine of the 13 configurations are feasible sometime during the course of the day. The figure also shows two optimal operating paths. The path traced by the broken dashed curve is for a day when conditions exclude the use of the cooling tower to provide free cooling in the water system. The dashed path is for conditions where waterside free cooling is possible. The conclusions which can be drawn from this figure are that there is always one operating configuration that is significantly more cost effective than any of the others; and, system loads change slowly enough that the optimal mode has an operating duration of at least two hours.

The curves also provide a comparison of the simplified algorithms and the comprehensive simulation of the water system. The open faced symbols of the curves are for the algorithms and the closed faced ones are for the simulation. The algorithms did an excellent job of predicting both the operating ranges and cost of the system configurations.

The next study illustrates that steam cost has little effect upon the optimal operating paths for the same set of load conditions as were used to generate 6.2. This study is shown in Figure 6.3. The results of reducing steam cost is to lower the cost curves of all configurations which require steam. A comparison of Figures 6.2 and 6.3 will show that a reduction of steam cost by a factor of three does not change the path traced by the optimal mode selection routine.

Similar studies based on the cooling and low heating load profiles are presented in Figures 6.4 and 6.5. The results of these studies are in agreement with those based on the high heating load profile.

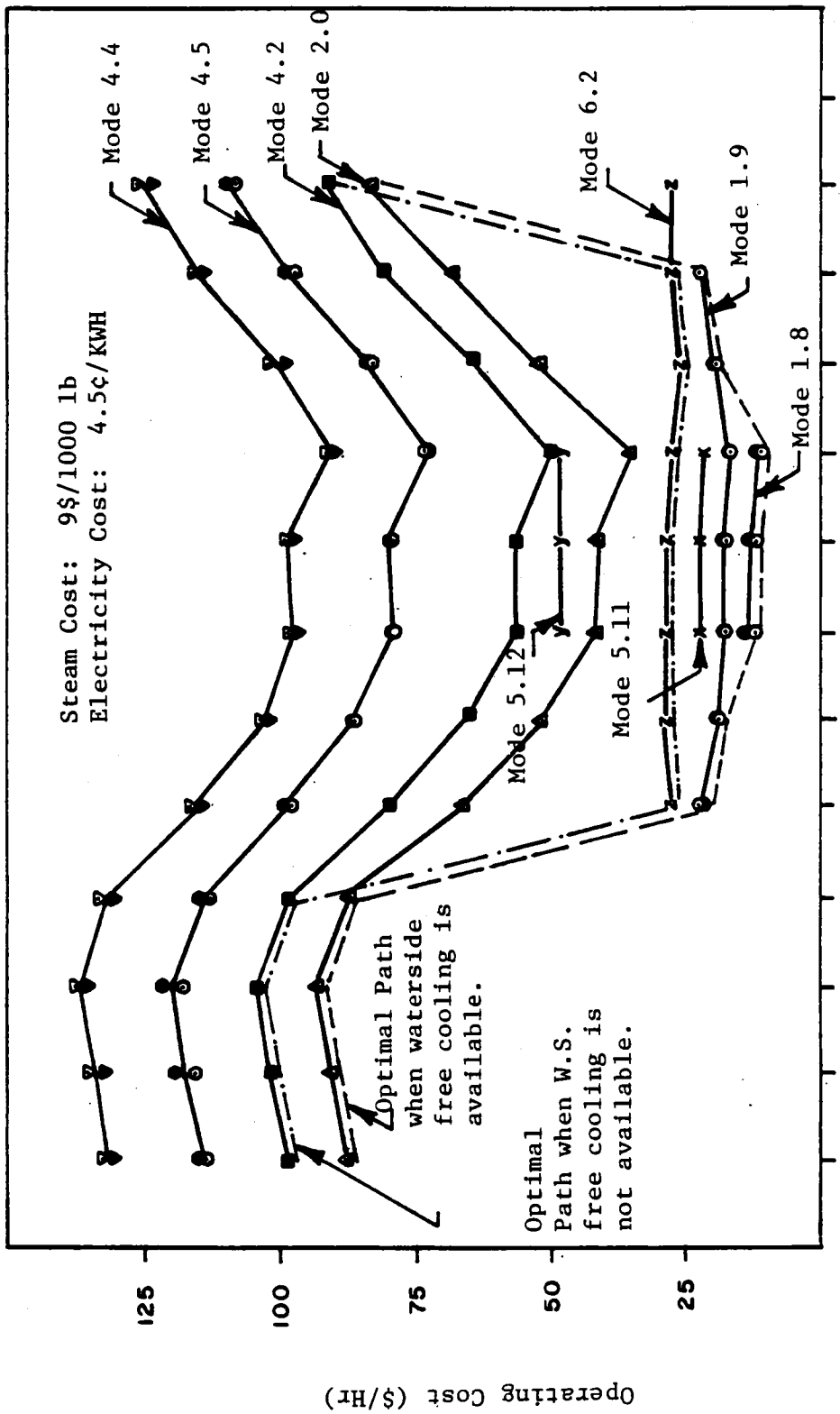


Figure 6.2. Cost and availability of system modes in response to the cooling and high heating load for a steam cost of $9/1000 lb. (Note the open symbols represent the reduced algorithm and the filled in symbols represent the comprehensive simulation).

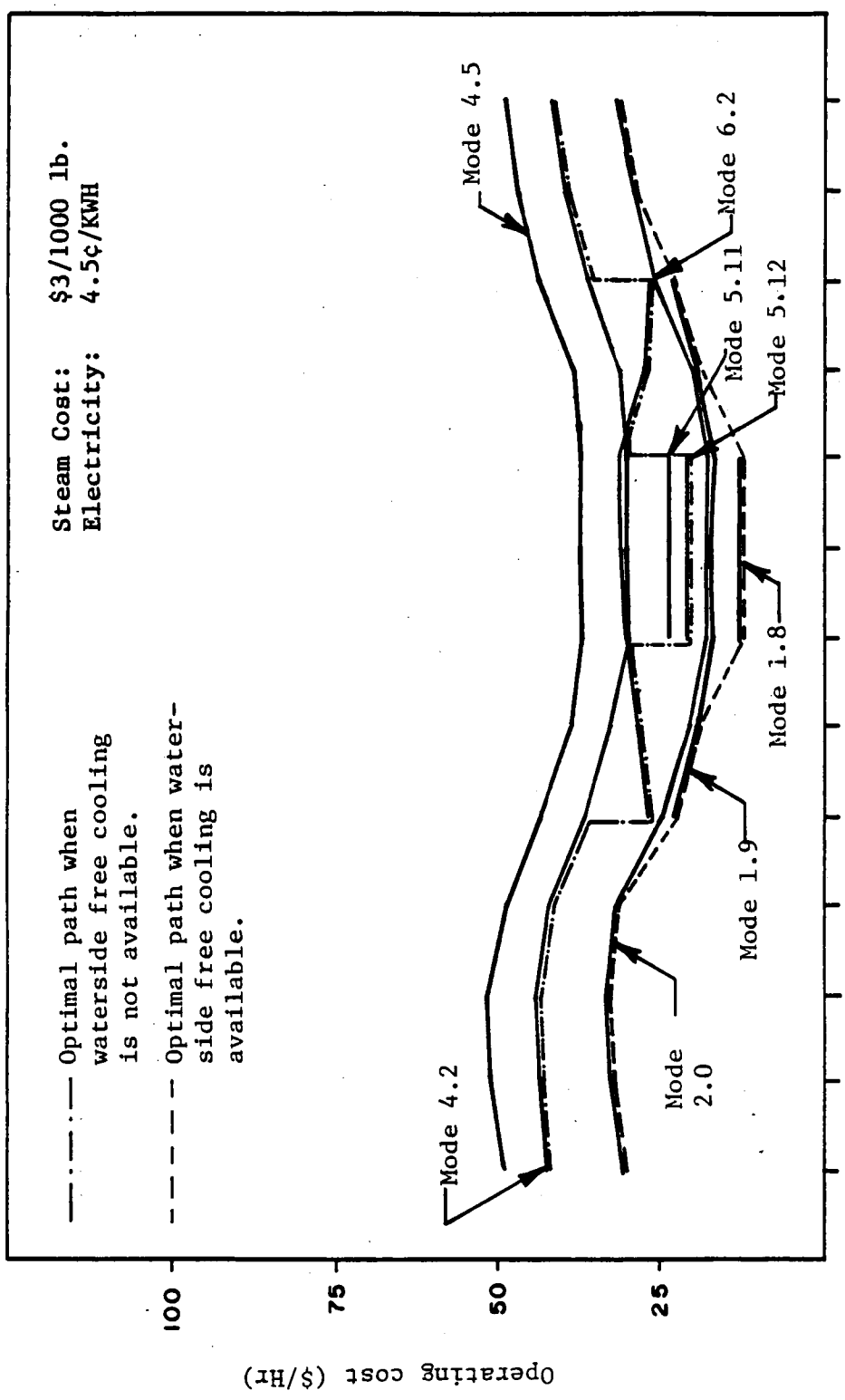


Figure 6.3. Cost and availability of system modes in response to the cooling and high heating load for a steam cost of $3/1000 lb.

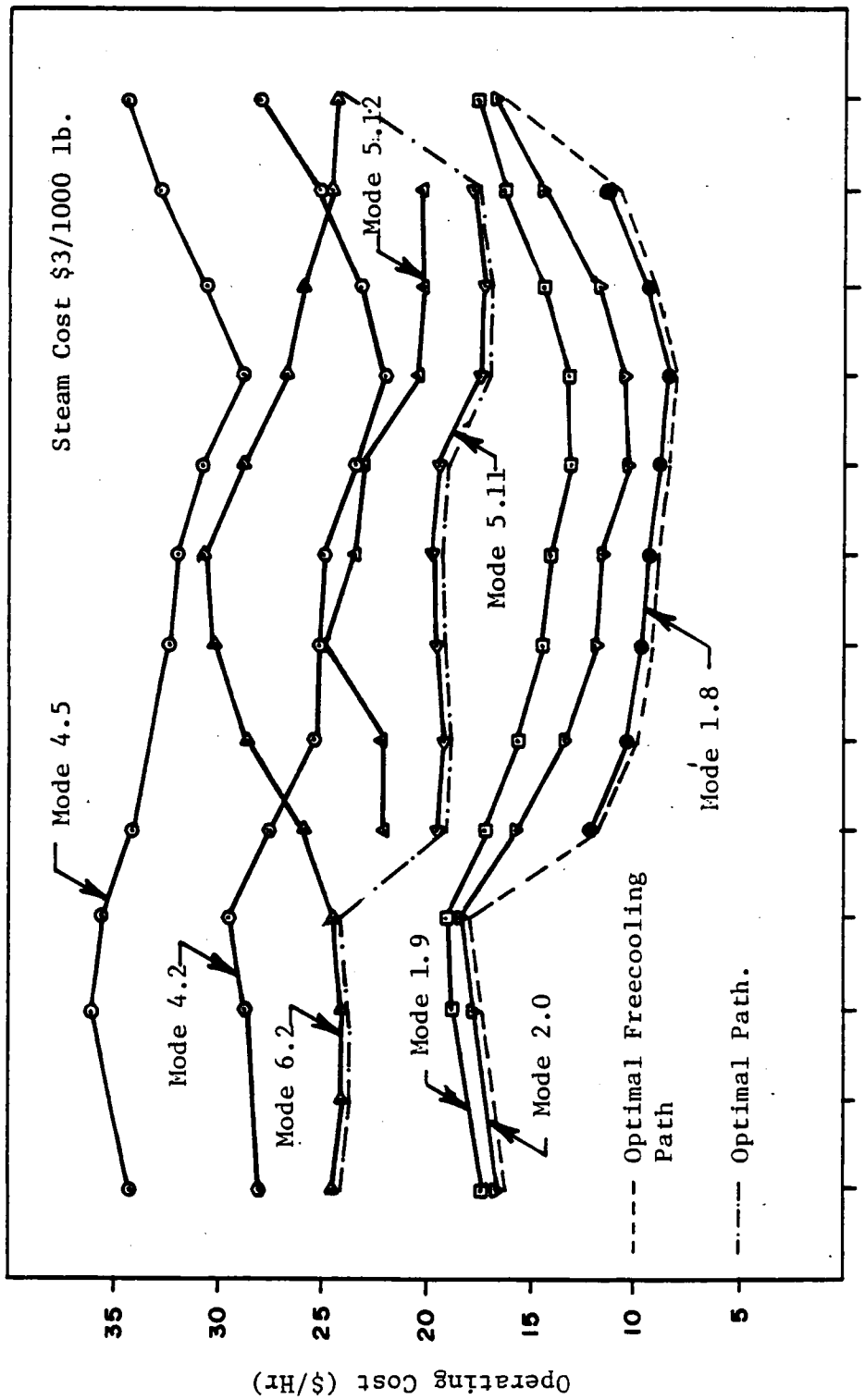


Figure 6.4. Cost and availability of system modes in response to the cooling and low heating load for a system of $3/1000 lb.

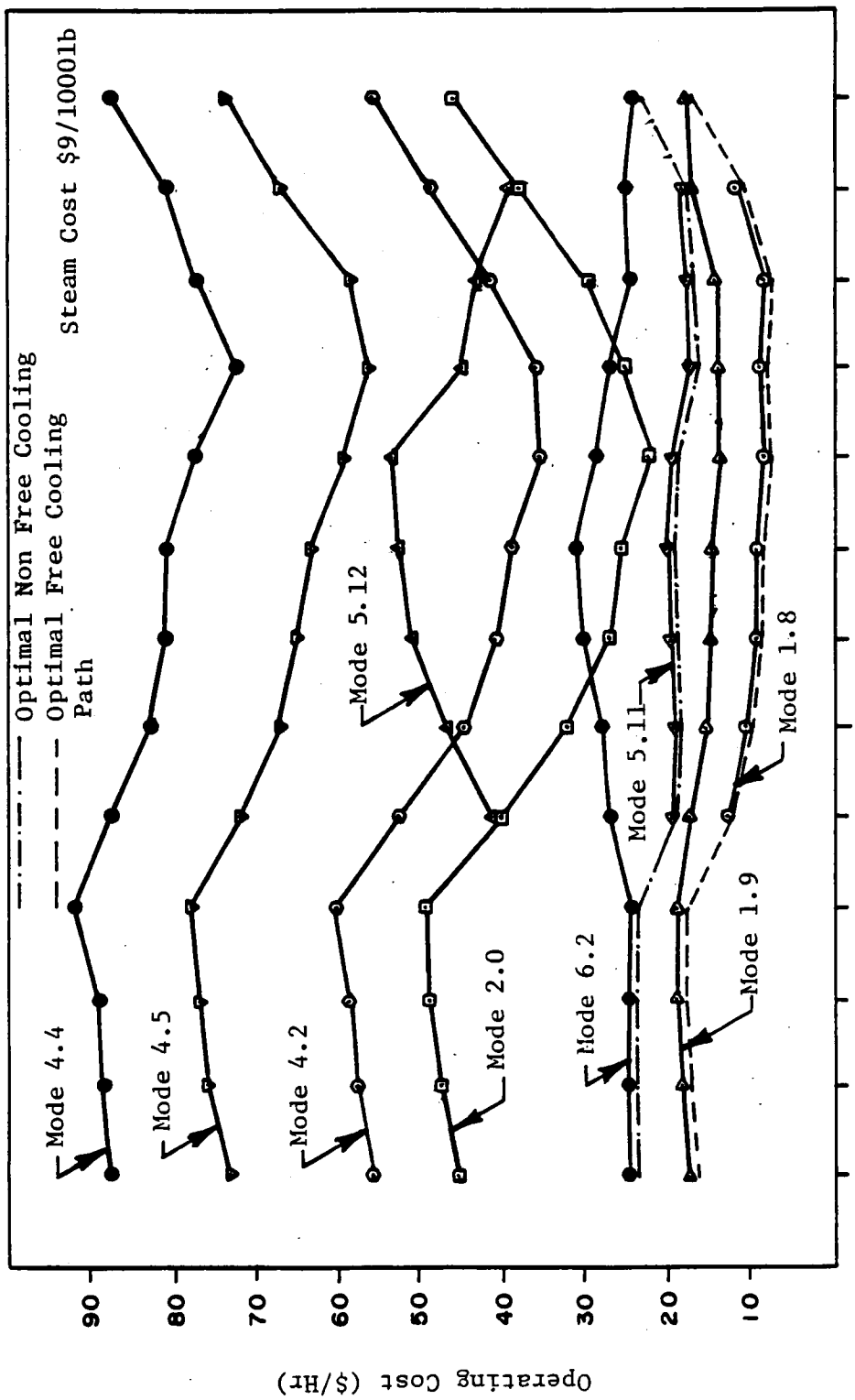


Figure 6.5. Cost and availability of system modes in response to the cooling and low heating load for a steam cost of $9/1000 lb.

The results which can be drawn from this are:

1. Recovered heat is generally more economical than steam heat.
2. If available, free cooling is more cost effective than conventional mechanical cooling.
3. Loads upon the water system change slowly enough that mode changes do not occur often.
4. The use of steam to provide mechanical refrigeration is not economical except under special circumstances (see discussion of peak load shedding).
5. There are distinct optimal operating paths which are easily selected using the simplified algorithms.
6. The simplified algorithms are an accurate and easily implemented tool which can be used in the optimization of the water systems operation.

So far this work has only addressed the operation of the primary water systems. In the next section, a comparison of the optimal operating paths with a straight economizer and a balanced heat recovery scheme will be made.

Comparison of Several Operating Strategies

In a building such as 590, where a high degree of operating flexibility is available, there are usually several operating strategies which will satisfy the system loads. When the various strategies involve only choices between water system configurations, the simple algorithms discussed earlier can be used to select the appropriate mode. But when outside conditions are such that free cooling is available through an economizer cycle, free cooling and pay cooling should be balanced to match the requirements of the building.

In this section, a comparison of three different operating strategies will be discussed. A straight economizer/steam heat strategy will be compared to a balanced heat recovery and an optimal mode selection strategy. The hypothetic load profiles will be the same as those used in the previous section. The optimal mode selection strategy assumes that both the building's heating and cooling loads are met by the water systems (the outside air damper in its minimum position). In the balanced heat recovery strategy a feedback control system similar to the one shown in Figure 6.6 is assumed to be manipulating the outside air damper to just match the cooling load on the heat pump to the building's heating requirements.

In Figure 6.7 are the first set of curves which show the relative cost of the three operating strategies. The high heating load profile from Figure 6.1 (page 132) is used in this study and steam is available at $9/1000 lbs ($10/MMBTU). From the figure one sees that the balanced heat recovery strategy is always more cost effective than either the economizer or the optimal operating path strategy (in this study it is assumed that conditions preclude the use of the cooling tower to provide free cooling on the water side). The cost of operation of the economizer cycle for this hypothetical day is $2358. The cost of the optimal operating path is $1598, while the balanced heat recovery is $1325. If free cooling is available from the cooling tower (see Figure 6.8) another optimal mode path is followed. The cost of this strategy for the day is $1369. It is obvious that when conditions are such that simultaneous free cooling is available on both the water and the air side of the system, there is very little difference between the two strategies.

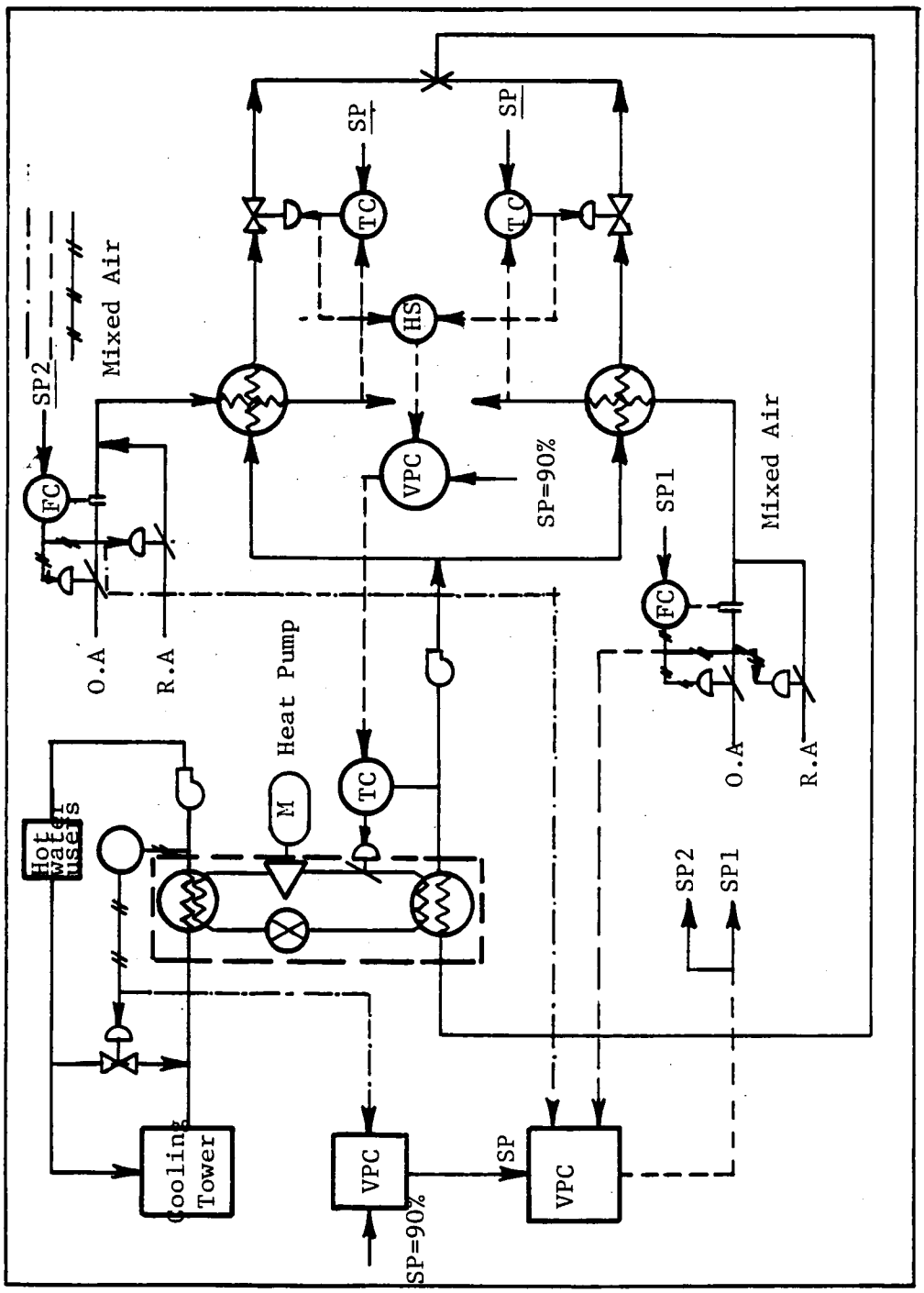


Figure 6.6. Balanced heat recovery with an economizer cycle.

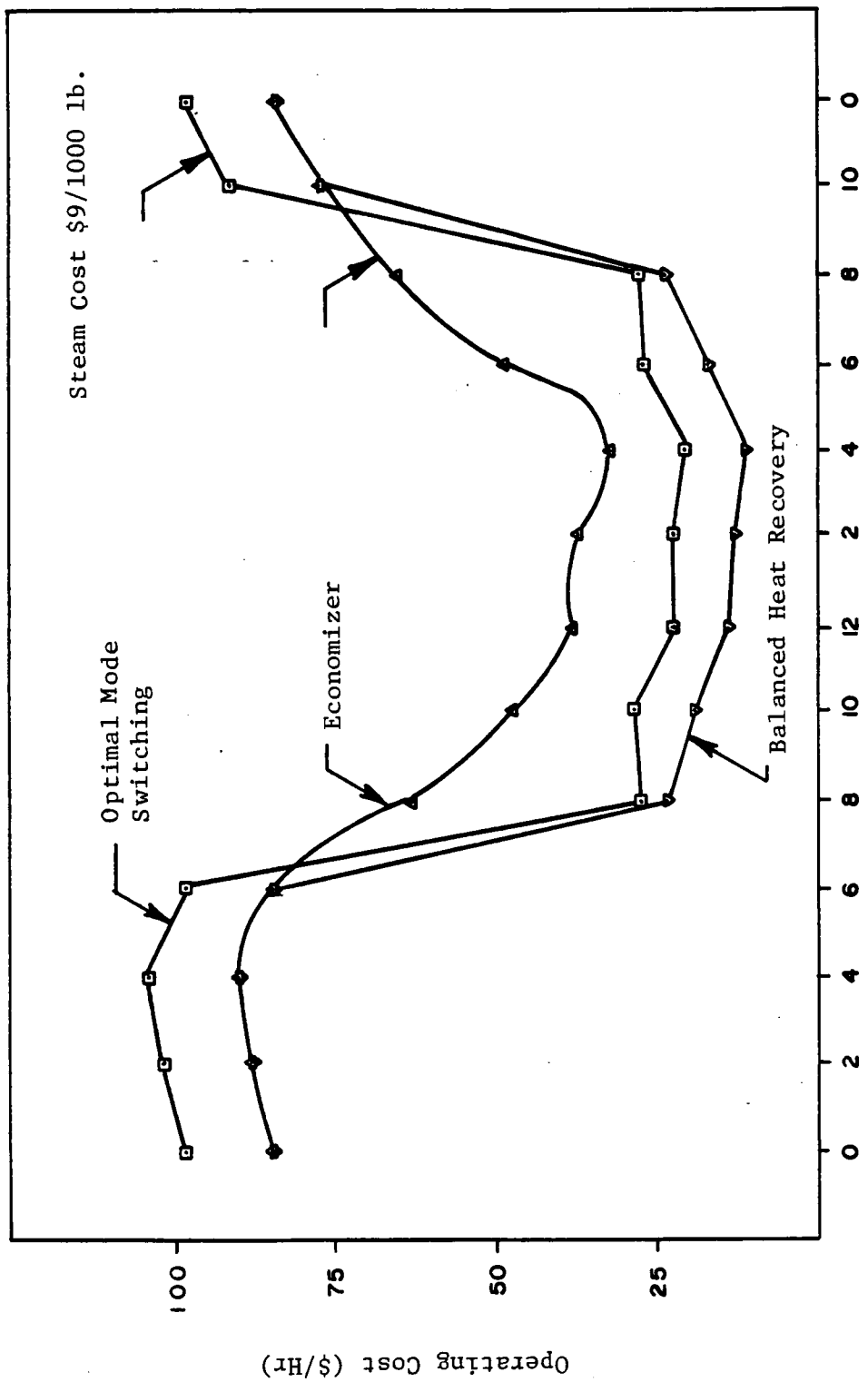


Figure 6.7. Comparison of the optimal mode path with the economizer and the balanced heat recovery schemes (high heat, no water side free cooling, and $9/1000 lb. steam).

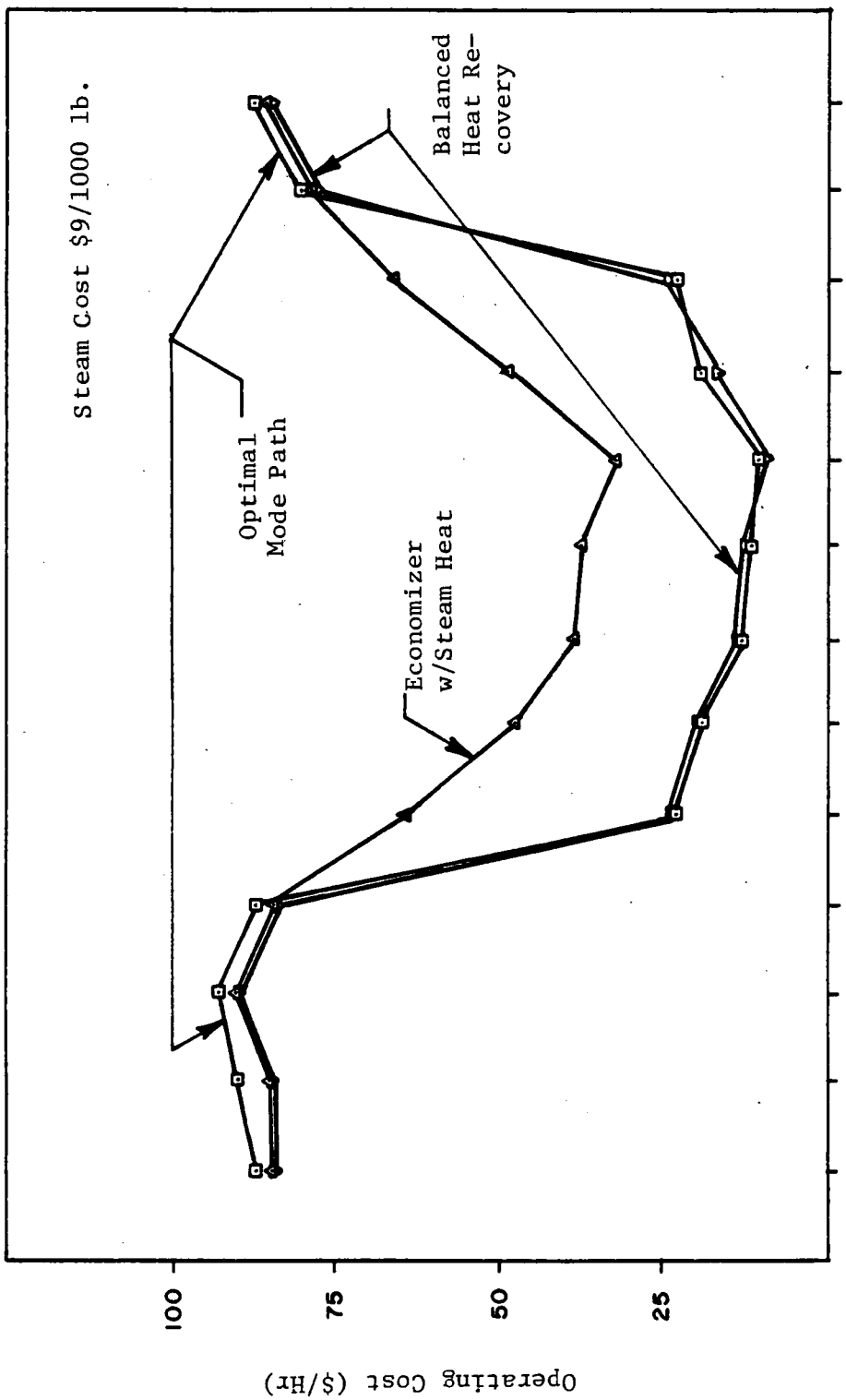


Figure 6.8. Comparison of optimal mode path with the economizer cycle and the balanced heat recovery schemes (high heat, water side free cooling, and $9/1000 lb. steam).

Figure 6.9 indicates that if the cost of steam is only $3/1000 lb. ($3.5/MMBTU) all three operating strategies are competitive if water side free cooling is available. The results of these studies are summarized in Table 6.1. The operating strategies for two heating load profiles and two steam rates are compared in Table 6.1. The cost reflects the purchased utilities for one day of operation. The results in Table 6.1 show that if steam is readily available and inexpensive, the economizer cycle is competitive with the balanced heat recovery and optimal mode strategy but if steam is available at a price significantly above $3.5/MMBTU the economizer cycle is not competitive. Figures 6.2 to 6.5 (pages 134-137) and Figures 6.6 to 6.13 were used to construct this table.

Peak Load Shedding

In this section the problem of peak demand management will be addressed. A method will be presented for the estimation of a target electrical peak at the beginning of the billing period of interest. This method relies upon the availability of a priori knowledge of the average load upon the mechanical equipment during the occupied hours of the period. The success of this proposed method depends on an accurate prediction of this load. The studies discussed here do not include specific methods for this prediction; however, it is common practice to relate system loads to the ambient outside air temperature. Several methods of predicting this temperature on a monthly basis are available (23). These predicted temperatures are generally presented in terms of frequency distributions. In other words, the monthly temperature is given in terms of the average temperature and its standard deviation. The number of hours that a specific temperature occurs during a month can be easily estimated

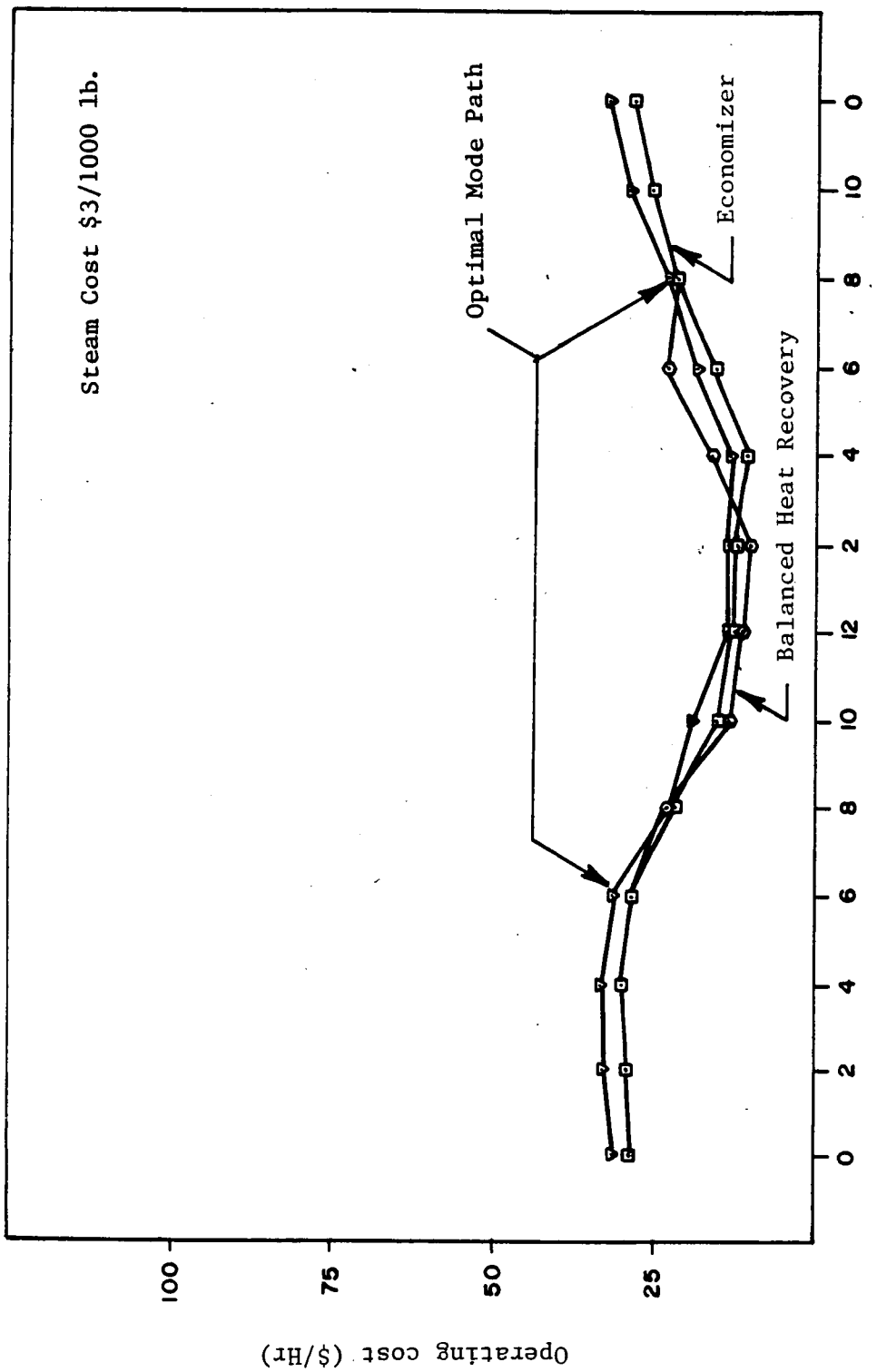


Figure 6.9. Comparison of the optimal mode path to the economizer and the balanced heat recovery schemes (high heat, water side free cooling, and $3/1000 lb. steam).

TABLE 6.1

COMPARISON OF THE ECONOMIZER WITH STEAM HEAT CYCLE TO THE OPTIMAL MODE SELECTION AND THE COMBINATION ECONOMIZER/HEAT RECOVERY CYCLES

	Steam Cost			
	3.5 $/MM BTU		10.5 $/MM BTU	
	Heating Profile			
Cycle	High	Low	High	Low
Economizer/Steam Heat ($)	786	300	2358	893
Economizer/Recovered Heat	810	323	1325	323
Optimal Mode with Water Side Free Cooling	895	345	1369	347
Optimal Mode Without Water Side Free Cooling	932	568	1598	568

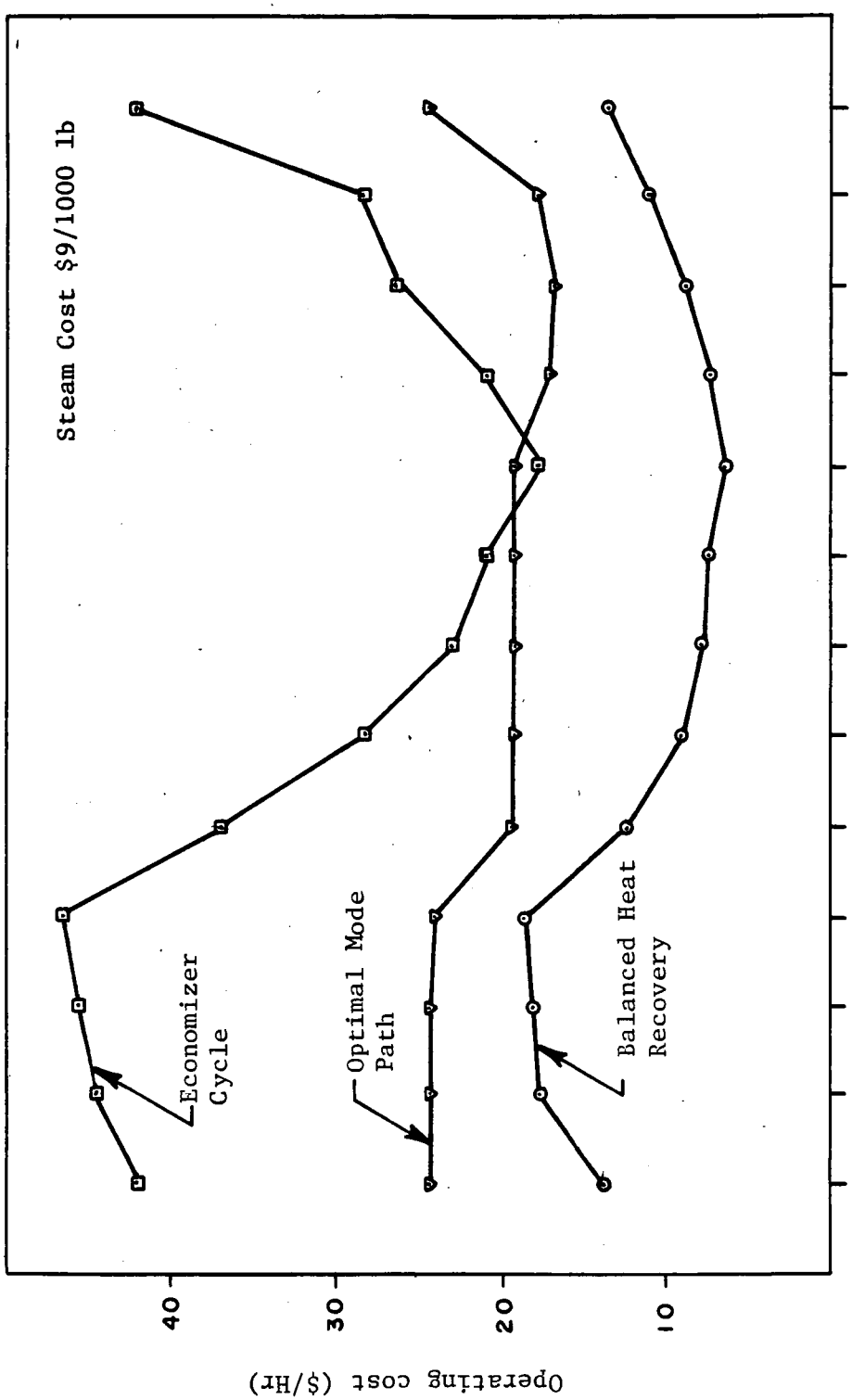


Figure 6.10. Comparison of the optimal mode path with the economizer and the balanced heat recovery schemes (low heat, no water side free cooling, and $9/1000 lb. steam).

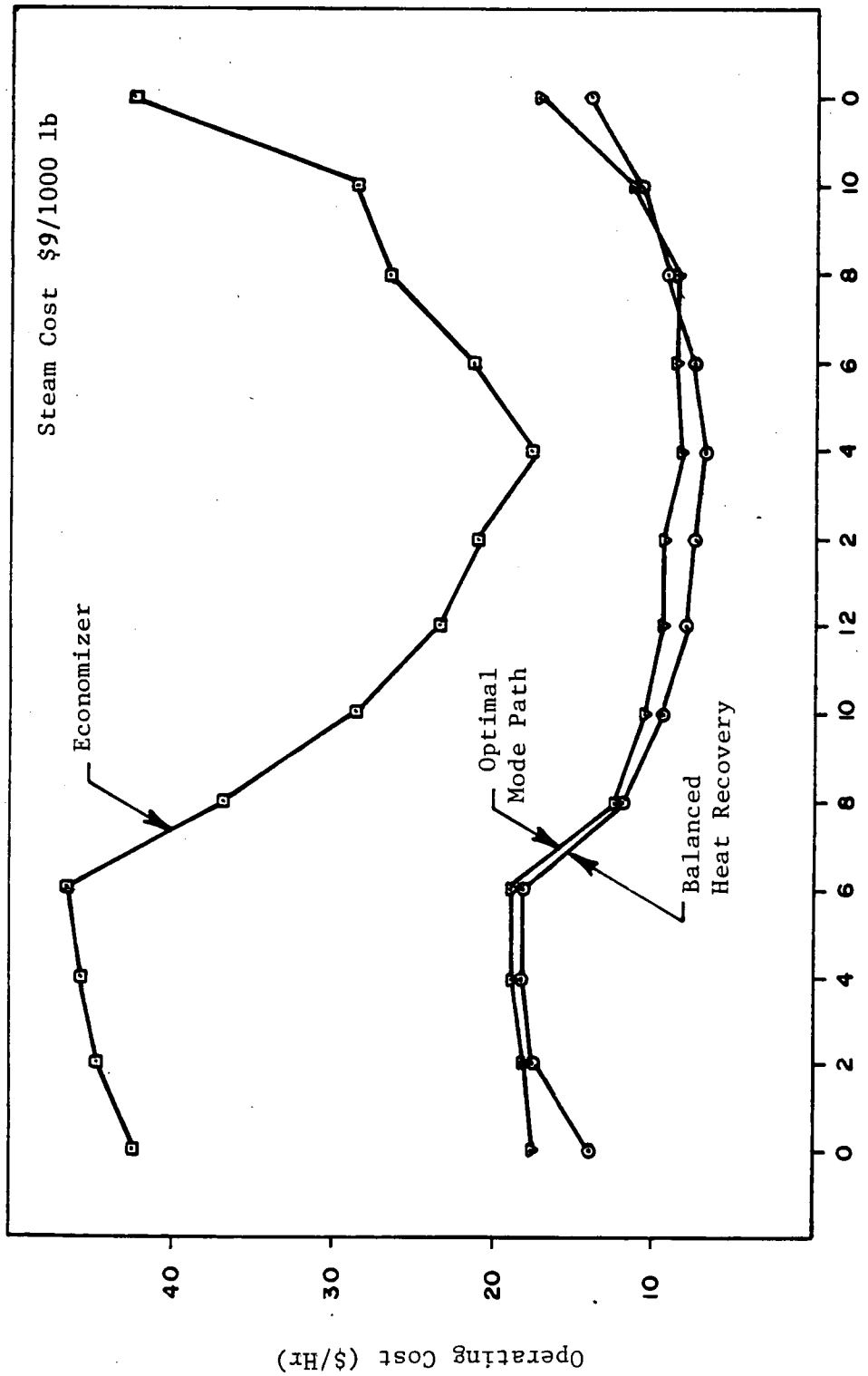


Figure 6.11. Comparison of the optimal mode path to the economizer and the balanced heat recovery schemes (low heat, water side free cooling, and $9/1000 lb. steam).

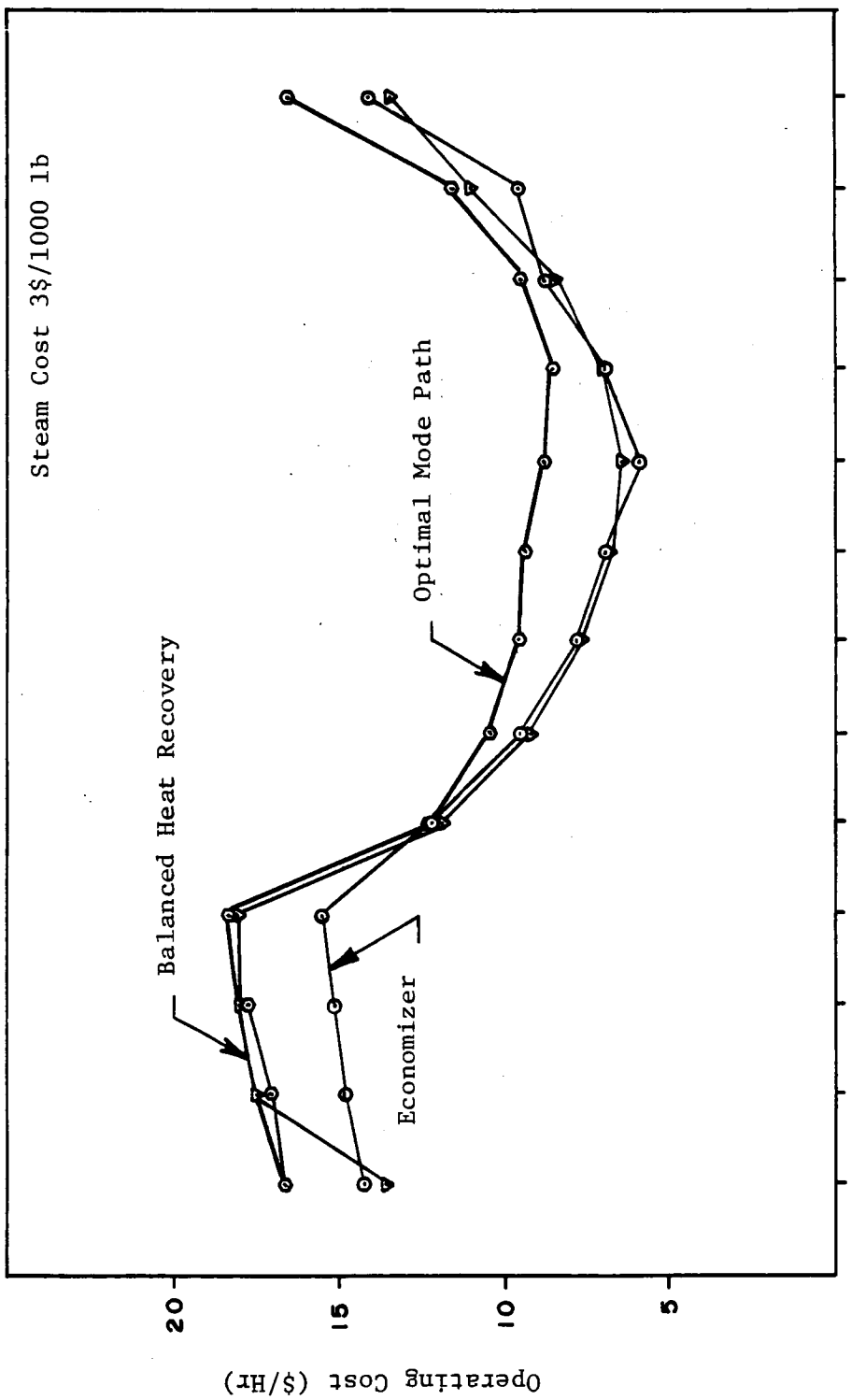


Figure 6.12. Comparison of the optimal mode path with the economizer and the balanced heat recovery schemes (low heat, water side free cooling, and $3/1000 lb. steam).

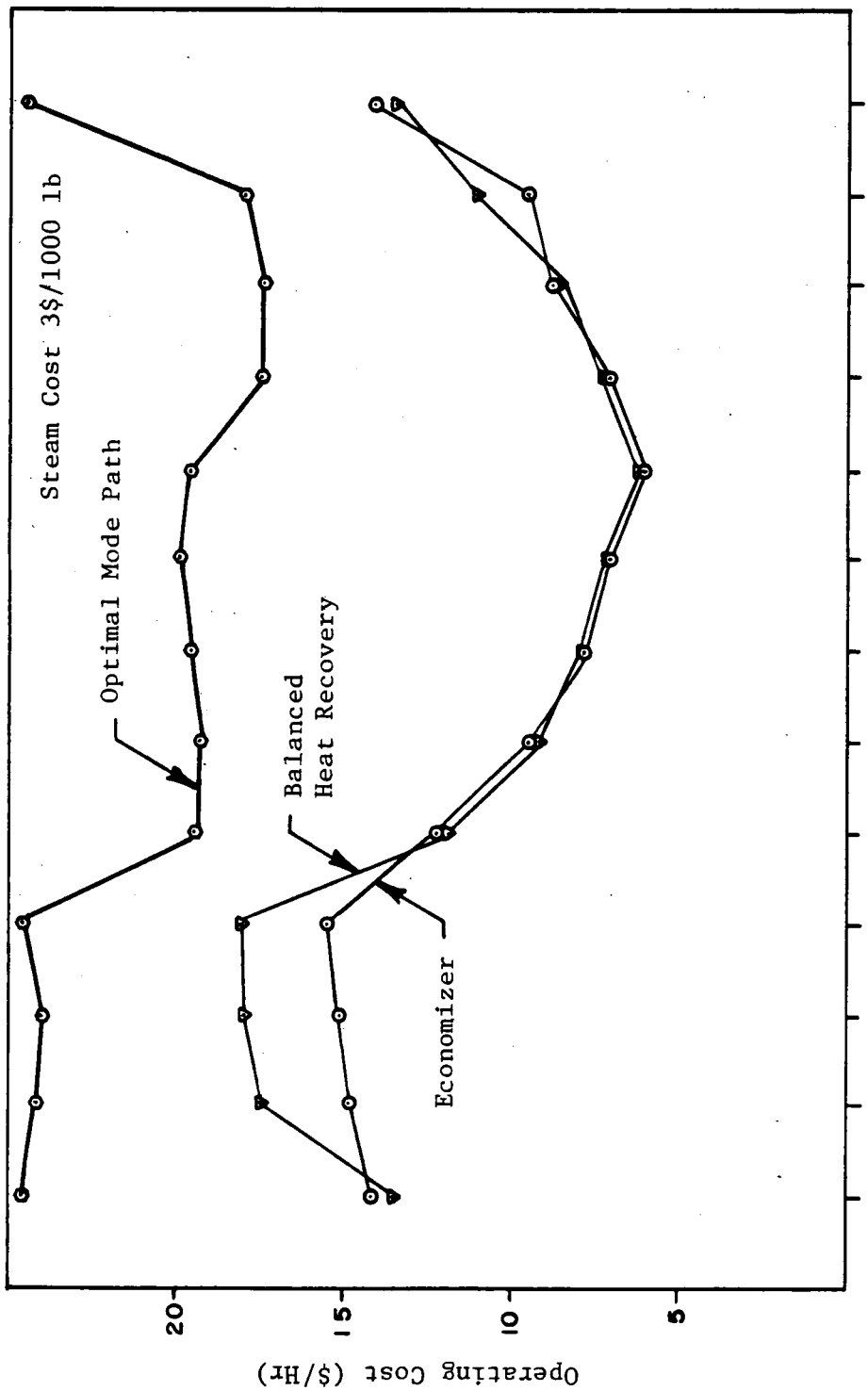


Figure 6.13. Comparison of the optimal mode path with the economizer and the balanced heat recovery schemes (low heat, no water side free cooling, and $3/1000 lb. steam).

by using the equation of a gaussian distribution. This equation is:

$$P(Tb) = \frac{1}{\sigma \sqrt{2\pi}} \exp \left[-\frac{1}{2} \left(\frac{T - Ta}{\sigma} \right)^2 \right]$$

where:

P(Tb) is the fraction of total hours Tb occurs

Ta is the average monthly temperature

σ is the standard deviation

By relating the mechanical equipment load to the outside air temperature, the load profile upon the equipment can be quickly computed by the familiar bell-shaped distribution. The average load upon the equipment will correspond to the average ambient temperature and 95 percent of the system loads will fall within plus or minus two sigma of the average load.

Thus a statistical approach may be used to predict the frequency of occurrence of system loads during the period of interest. From this prediction a procedure can be developed for estimating the target electrical peak. This peak becomes the system load where the steam chiller is used to limit electrical demand. The rest of this narrative will be concerned with the optimum utilization of the steam chiller to limit electrical demand and the selection of a target electrical peak.

Operating Cost Versus Peak Demand Cost Optimization

One approach to optimizing peak versus operating cost is to run the large steam chiller whenever the building's electrical demand approaches a critical target peak. The standard approach is to turn the large electric chiller off and start the large steam chiller. The electrical

consumption in the water systems is reduced significantly by taking this option, but the cost of operation increases significantly. A preferred approach is to operate both large chillers in parallel. The load on the large electric chiller is modulated to maintain the electrical demand below the peak.

A sample steady-state peak demand limiting control system is shown in Figure 6.14. The demand controller is provided with the total system electrical demand and the value of the target peak. The controller modulates the flow distribution between the two chillers to shed electrical demand at the chiller. As the system load increases, the steam chiller meets more and more of the load and thus maintains demand within the boundary set by the target peak.

If the peak demand is to be managed effectively, it is essential to establish the proper target peak at the beginning of the billing period. If the target peak is initially set too low and has to be updated, then the total cost of operation for the billing period is increased due to excess steam usage. If the target is set too high, the operating cost increases due to an unnecessarily high peak demand charge. The purpose of this work is to illustrate the effect of the target peak upon the total system cost.

Determination of the Optimum Target Peak

In Figure 6.15 is a hypothetical load distribution. The distribution is simply a summary of the frequency that a given load occurs during the period of interest. For instance, the average load occurred 8 percent of the time. The total hours that this load was imposed upon the system can be computed as (.08 X total hours). Since each load occurs with essentially the same frequency, this distribution represents a period when outside

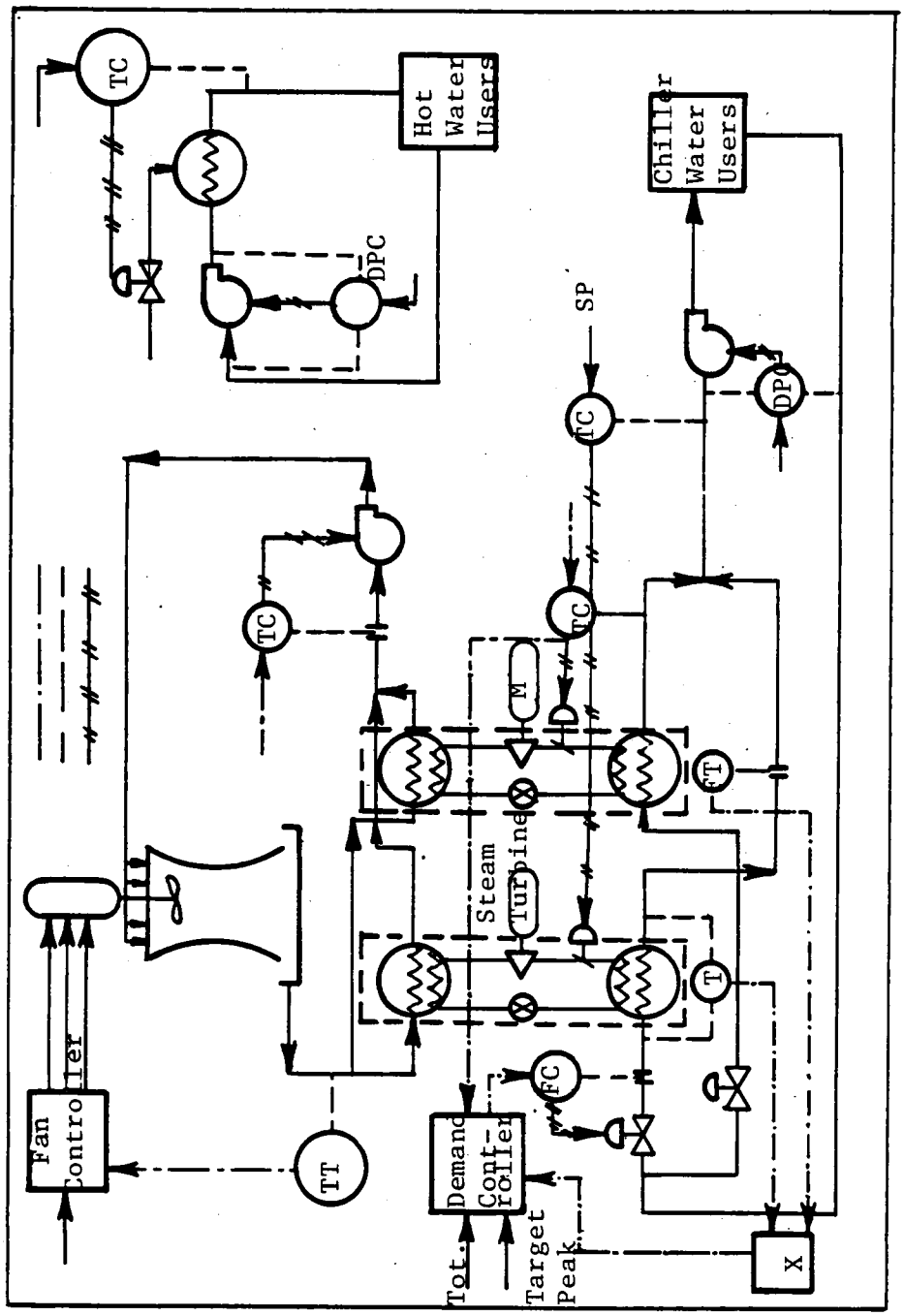


Figure 6.14. Peak load shedding control system.

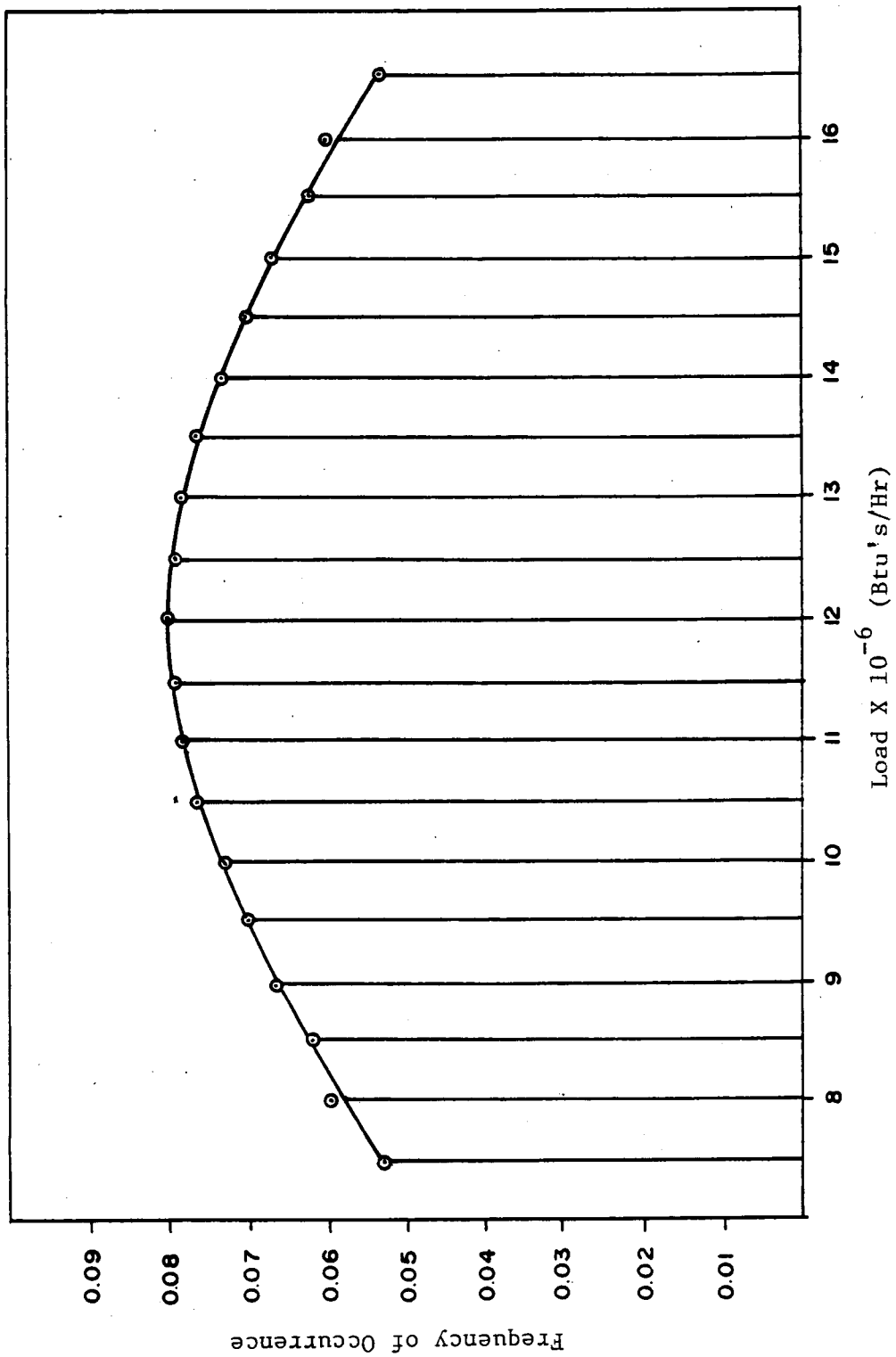


Figure 6.15. Frequency of occurrence of system loads (average load 12 million BTU's/hr, standard deviation 5.0).

conditions changed significantly over the span of the period. This could easily represent the occupied hours from September to December. This frequency distribution will be used to illustrate how the selection of the target peak affects the total system cost. If the target peak is chosen to be 12.0 million BTU's/hour (the mean load), then approximately 50% of the time the steam chiller will be used to limit electrical demand.

In Figure 6.16 are a set of curves which show the operating cost for the billing period as the target peak is varied. Four curves are plotted in the figure. On the right ordinate are values of Beta. The target peak is given as a function of Beta and the average load of the distribution profile. The expression relating the target peak to Beta is:

$$Tp = M + BM$$

where:

Tp is the target peak

M is the average daily load (12 million in this example)

B is an operator which relates the integral of the frequency distribution curve to the selected target peak.

The fraction of the time that the system load exceeds the target peak is determined by selecting the value of Beta and reading the abcissa of the Beta curve. At that value of Beta, the total cost and the two component costs which this cost is comprised of can be read from the figure.

The average hourly operating cost curve is the purchase cost of steam and electricity for the period divided by the total hours in the period. The total charge for the period is computed as the sum of the average peak demand charge and the average purchased utilities charge times the number of hours in the period. If the target peak is set at the

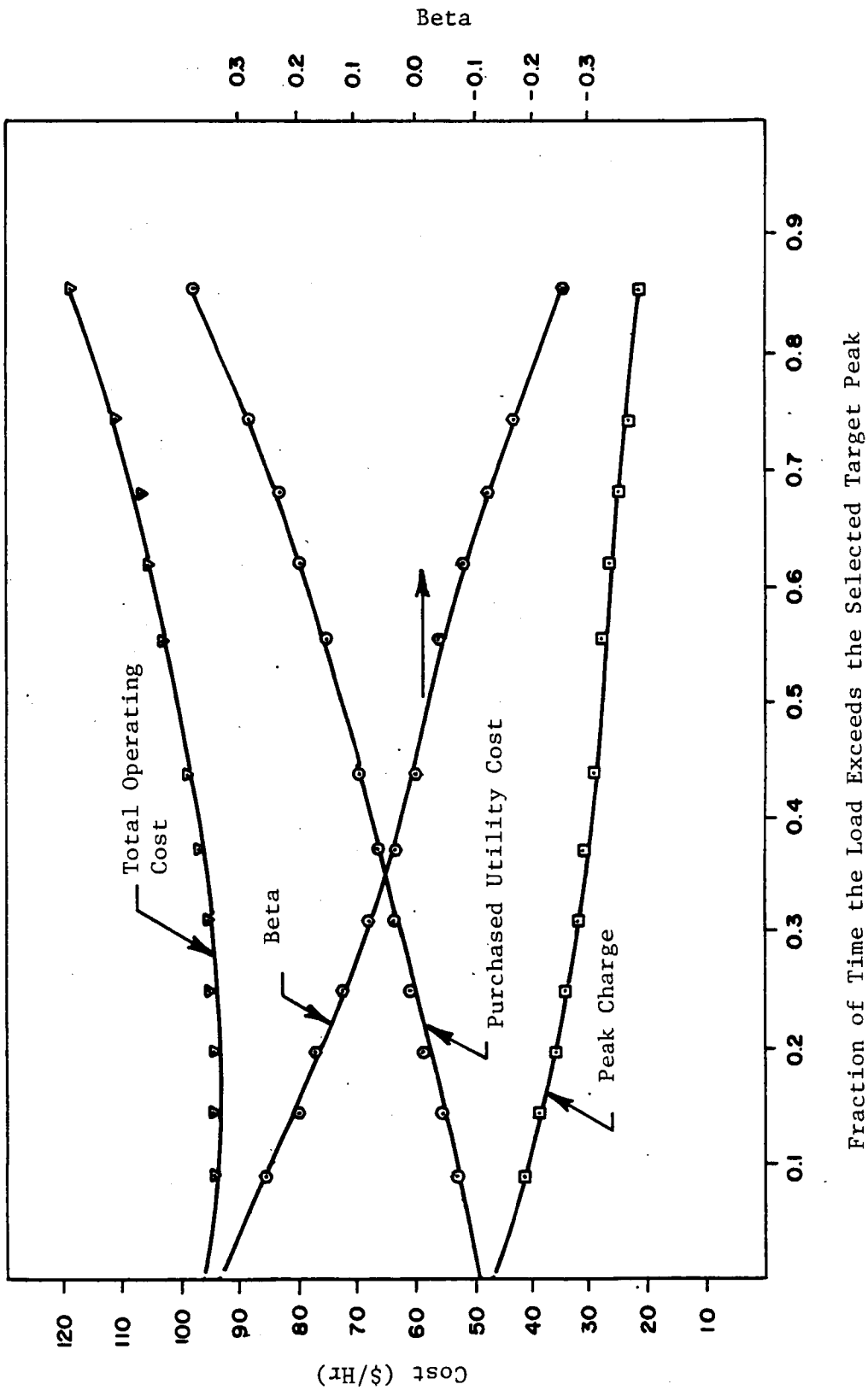


Figure 6.16. System cost as a function of the selected target peak (based upon the distribution of Figure 6.15, peak demand limiting control scheme of Figure 6.14 is used whenever the load exceeds the target peak).

average load (B = 0), the loads on the system exceed the target peak 40% of the time. For a three month period (720 hours of occupied operation), the peak charge is $20,160 and the purchased utility cost is $49,680. The total cost of operation during the occupied hours is $69,840 for an average hourly operating cost of $97/hr. (For this study the peak demand charge is $20/KWH.) From the total average hourly cost curve, the optimum target peak occurs when the system load exceeds the target peak 15% of the time. This corresponds to a Beta value of 0.2.

In Figure 6.17 are a set of curves for a study where the large electric chiller is used in parallel with the small electric chiller to shed excess demand. One readily sees that this strategy is never cost effective.

And finally, in Figure 6.18 are a series of curves which illustrate the effect of the peak demand charge upon total average operating cost for the preferred demand limiting strategy. This figure indicates that the peak charge would have to increase to approximately $70/KWH before the results of this study are significantly altered.

The conclusions which can be drawn are:

1. For any demand limiting scheme, the steam chiller should only be used to meet those loads which exceed the target peak.
2. The target peak load can be safely estimated by determining the average daily load on the water systems during the period of interest. Thus, T_p is computed as TP = M + 0.15 X M.
3. A target peak set too low is much more damaging than one set too high.

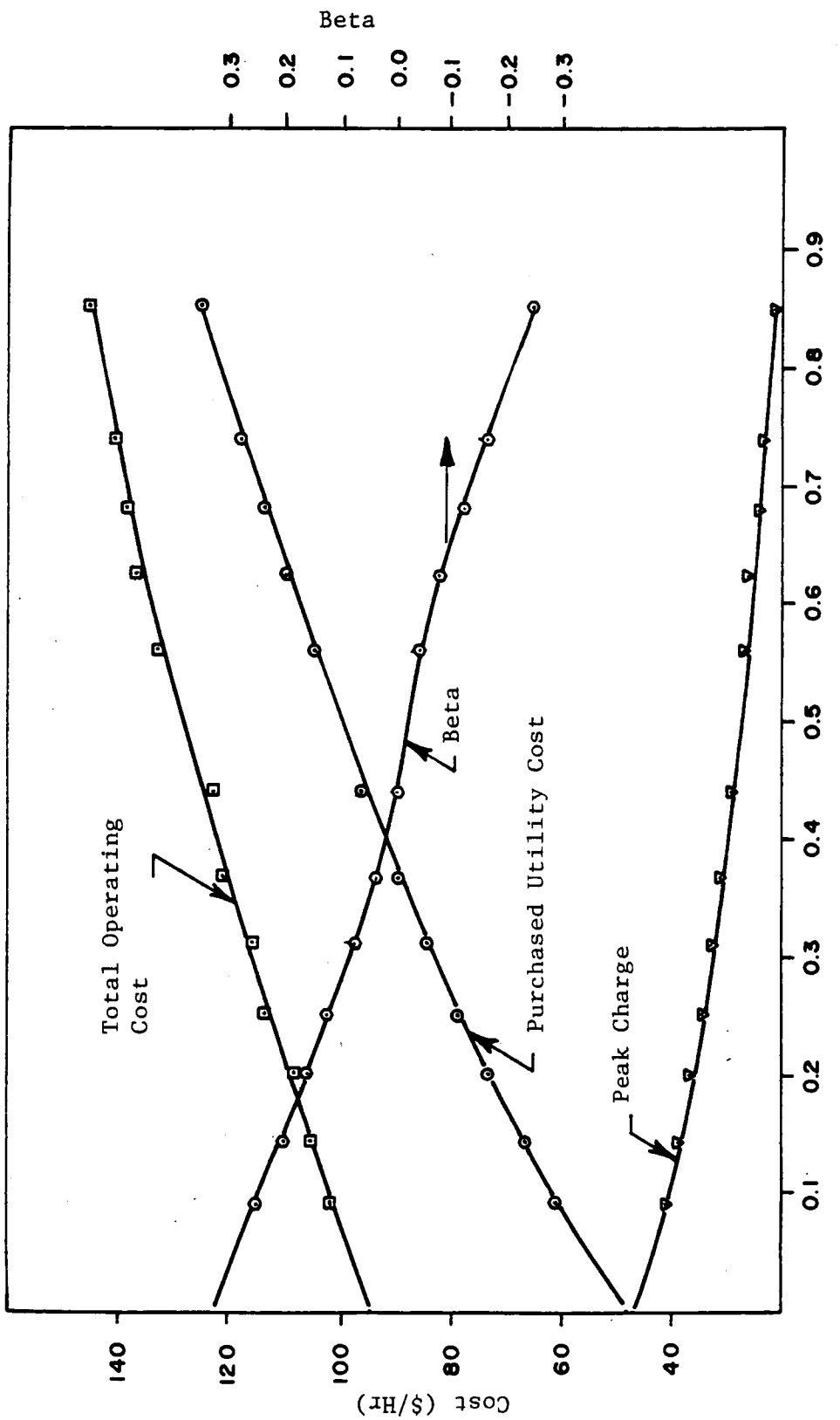


Figure 6.17. System cost as a function of the selected target peak (based upon distribution of Figure 6.15, peak demand limiting control scheme uses Mode 3.5 whenever load exceeds the target peak).

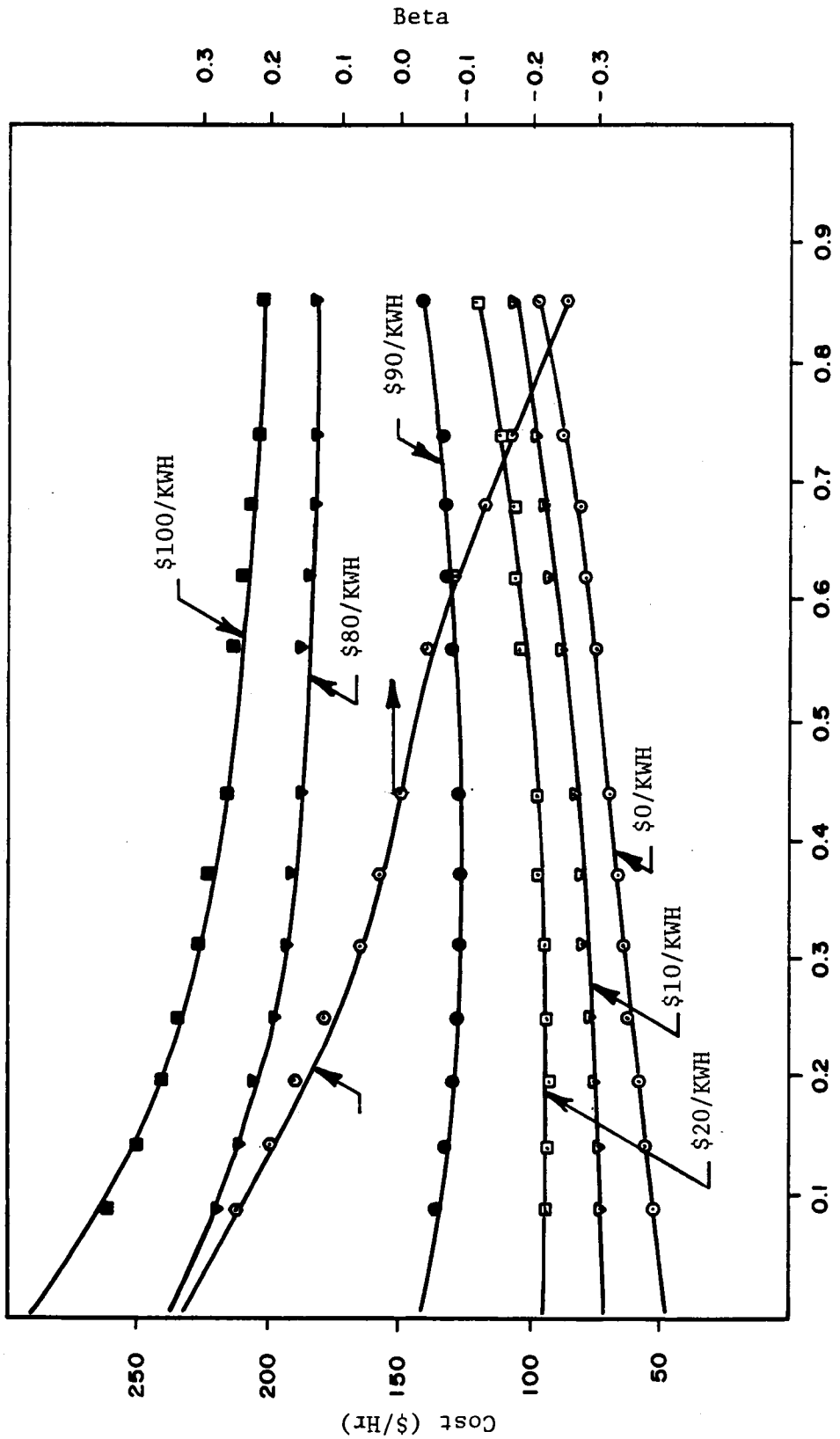


Figure 6.18. Effect of peak demand charge upon system operating cost (based upon the distribution of Figure 6.15).

Cooling Tower Supply Temperature Study

In this section the effect of the available cooling tower supply temperature upon the operating cost of several water system modes is discussed. The intent is to show that the cost of steam will have to change significantly from the mid-1979 rate before a mechanical refrigeration with steam heat mode becomes more cost effective than a heat recovery one.

In Figure 6.19 is the first of three studies which were conducted. The cost curves of this figure were generated by establishing the heating load, cooling load, hot water supply temperature, and chilled water supply temperature, then letting the tower supply temperature vary from 90 to 55°F. Three cost curves for Mode 4.2 are shown. The curves are for $9/1000 lbs. steam ($10.5/MMBTU), $6/1000 lbs. and free steam. The cost of electricity remains constant at 4.5¢/KWH. The cost of two heat recovery modes are also shown. Mode 6.2 has the large electric chiller supplying all the heating and cooling with any excess heat being rejected by the tower. Mode 5.11 has the large electric chiller supplying whatever cooling the heat pump does not provide.

For the conditions in Figure 6.19, the cost of steam has to be approximately $2.5/1000 lbs. before there is a supply temperature where Mode 4.2 becomes more cost effective than either Modes 6.2 or 5.11. For the $2.5/1000 lbs. steam the cross over temperature is 70°F.

Figures 6.20 and 6.21 present studies for cooling loads of 500 and 750 tons. The heating load is held constant in both of these studies. The results agree with those presented in Figure 6.21.

The conclusion for 590, at the mid-1979 utility rates, is that recovered heat is always more economical than steam heat. It should be noted

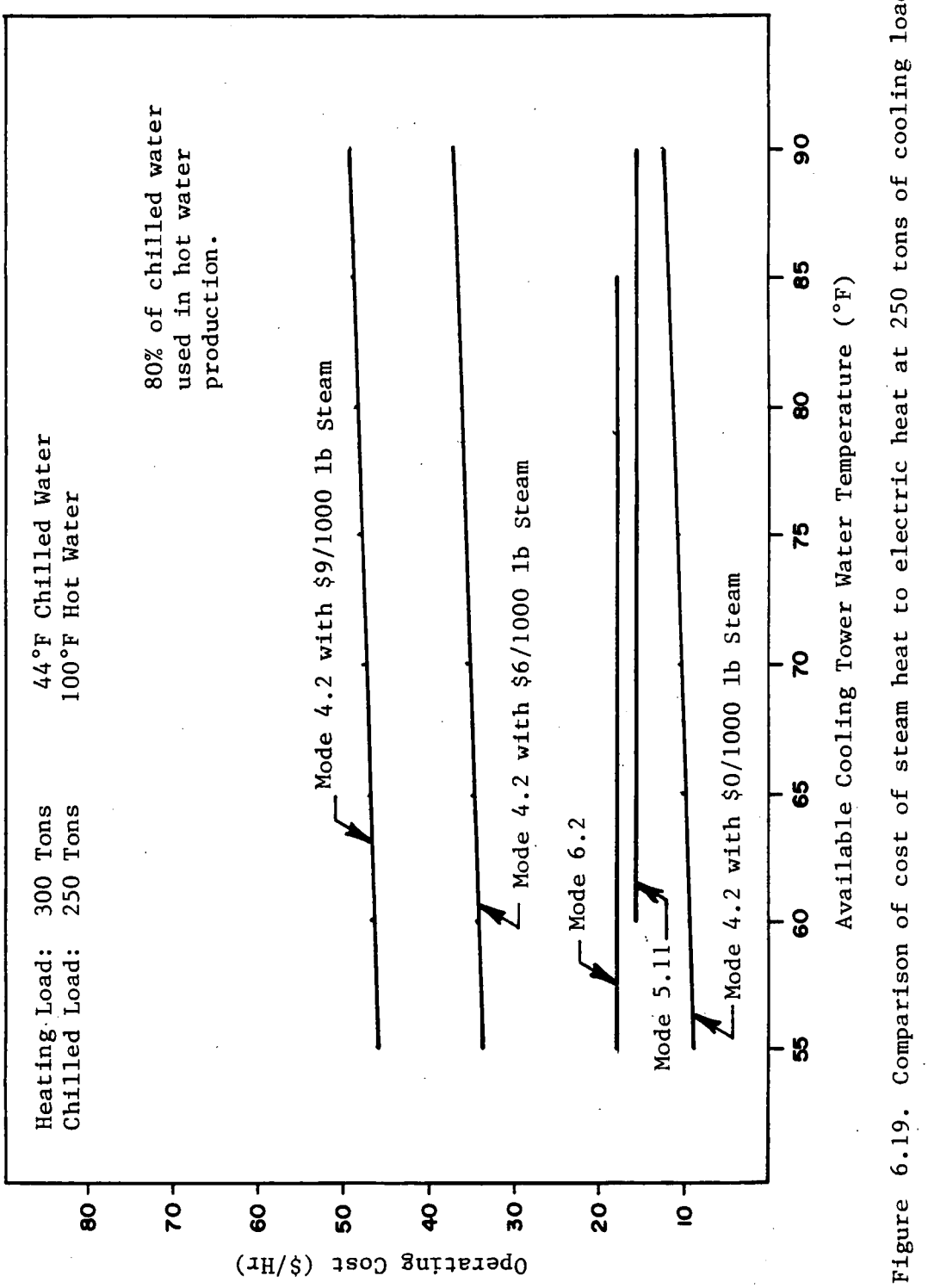


Figure 6.19. Comparison of cost of steam heat to electric heat at 250 tons of cooling load.

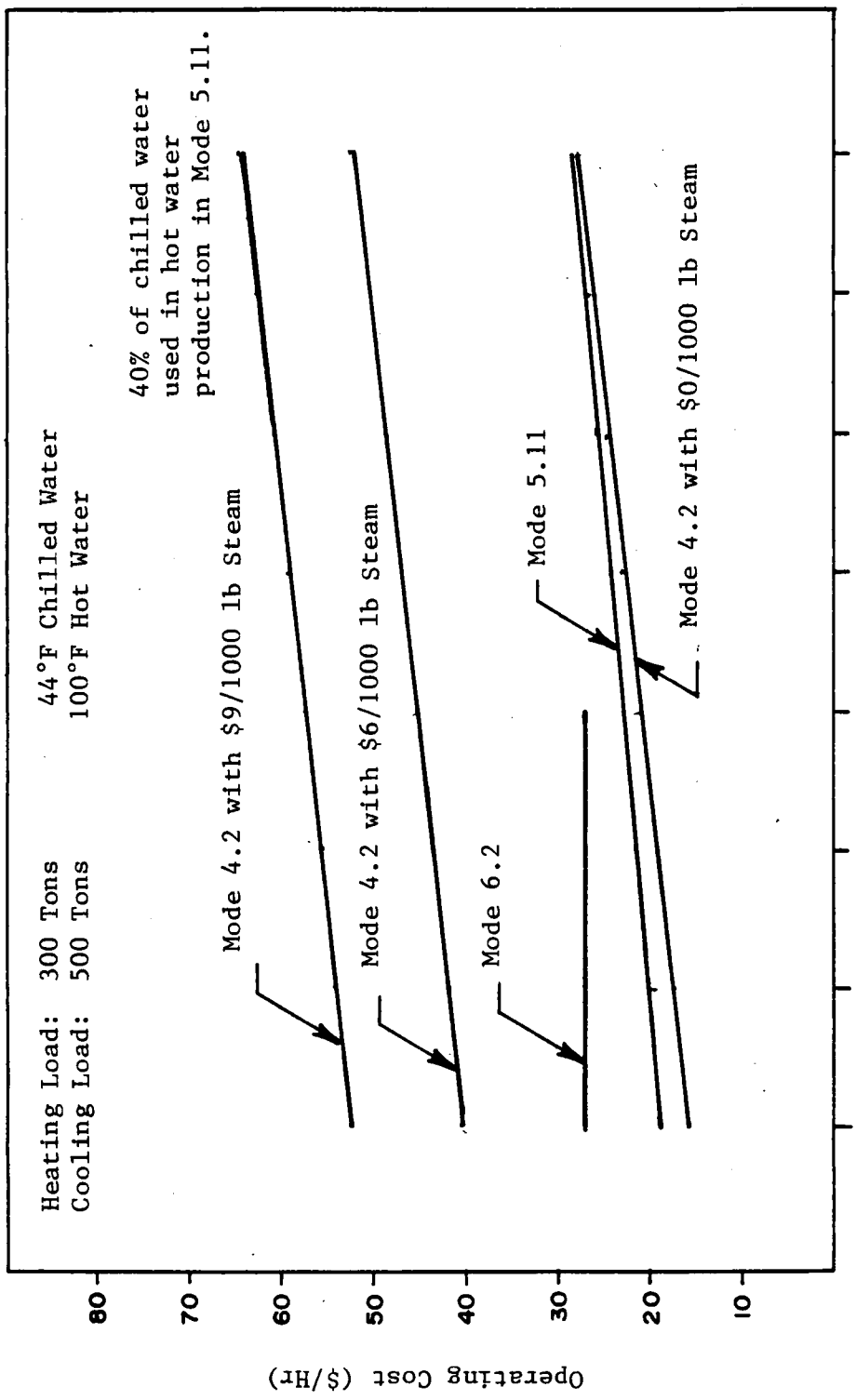


Figure 6.20. Comparison of steam heat to electric (recovered) heat at 500 tons of cooling load.

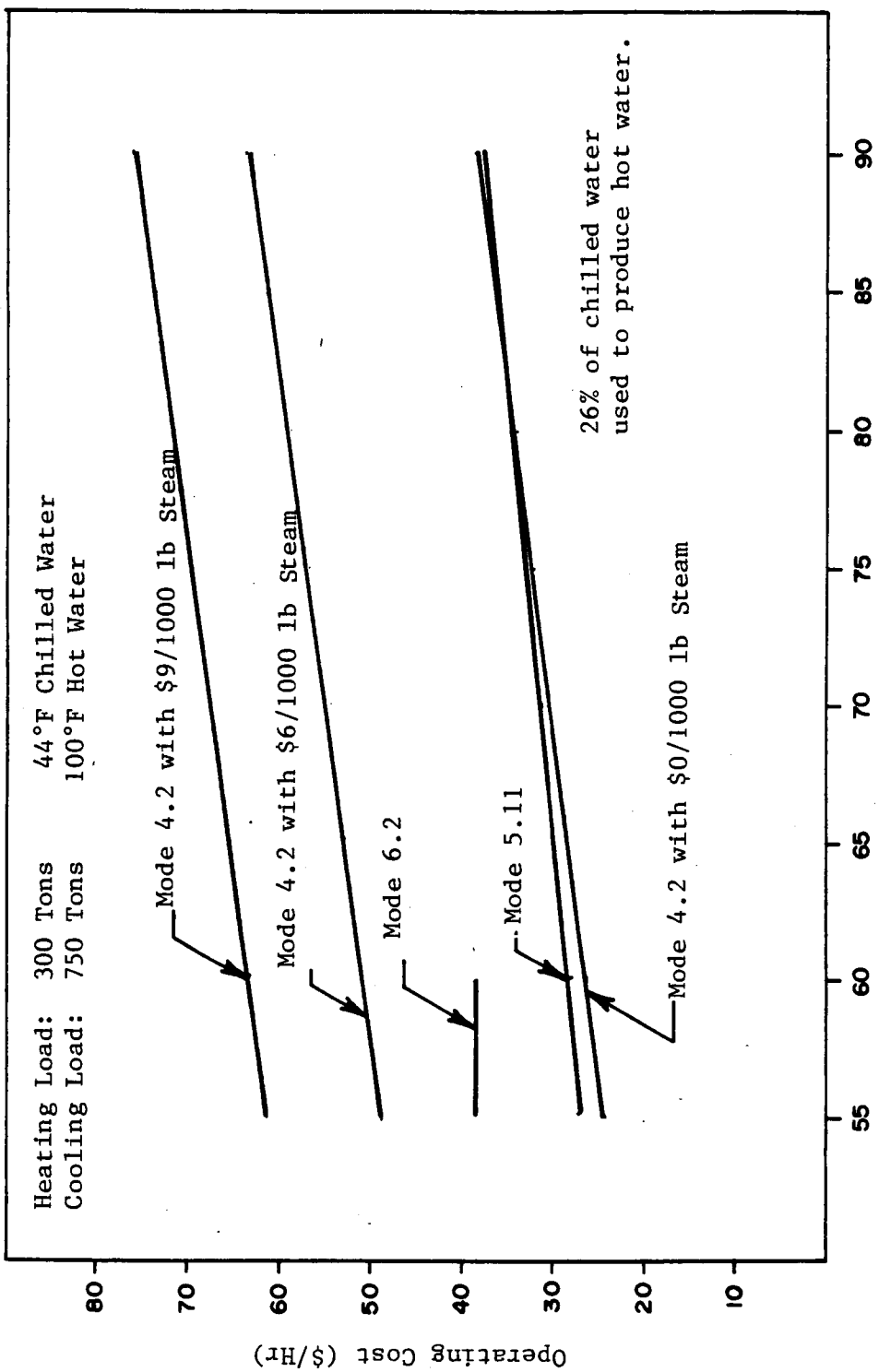


Figure 6.21. Comparison of steam heat to electric (recovered) heat at 750 tons of cooling load.

however, that the simplified algorithms developed will always select the most effective mode and this discussion is just for reference.

Concluding Remarks

The purpose of this chapter has been to address the problem of operating the total HVAC system in a cost effective way. Three case studies were presented which contrast current operating strategies to those which arise if realistic energy and modern equipment designs are factored into the analysis. The results from these studies suggest that the current trend, towards systems that balance the initial capital investment against the lifetime operating cost, is justified. The major conclusion which can be drawn from these studies is that, at current utility rates, electricity is preferred to steam for meeting the building's heating and cooling loads if heat recovery devices are available.

CHAPTER VII

CONCLUSIONS

There are two primary conclusions which may be drawn from the work presented in this thesis. The first one is that computer based simulations are effective tools in a priori studies of systems which are not yet available for on-line study. Though this work focuses upon a specific system, it does demonstrate a systematic approach to a computer based analysis of a complex plant.

The second conclusion concerns the specific type of system studied, which in this work is a large office building's nonconventional and very versatile HVAC system. The purpose of the study was to understand the interactions between the various subsystems of the building and to develop an optimization strategy suitable for an online implementation. These objectives were achieved and presented in terms of sensivitity studies and straightforward heuristic algorithms for the energy efficient operation of the building. The results presented in this work will eventually be tested against the actual building operation. However, these results are presented with a high degree of confidence that they will, in general, be consistent with actual plant operations.

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VITA

John Allen Arnold was born in Memphis, Tennessee on April 2, 1954. He graduated from Memphis Melrose High School in June of 1972. The following September he entered The University of Tennessee, Knoxville and in March of 1978 he received a Bachelor of Science degree in Chemical Engineering. During the years between 1972 and his graduation, he participated in the engineering Co-op program and was employed by the Tennessee Eastman Company in Kingsport, Tennessee.

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